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(54) Abstract Title
CVT with planet members connected to a link which maintains a circumferential connection

(57) A continuously variable transmission comprises planet members 21 connected to planet carrier arms 27 by a link or trailing arm 26 which allows a radial position of the planet members 21 to vary in response to a variation in axial separation of outer and inner races 12, 35 while maintaining a circumferential connection. The outer and inner races 12, 35 each comprise axially spaced relatively axially movable parts 13, 14 36, 37 controlled by ball screws 15; 38 driven by an electric motor. Planet members 21 comprise of two shells 22, 23 having a surface of revolution defined by a curvilinear generatrix that engages with respective parts of the two races 12, 35. Shells 22, 23 are joined to the link 26 by an intermediate element having needle roller bearings 25. In other embodiments the planetary members have a more pronounced prolate shape (figs 4 and 5) and backlash damping may be provided for the ball screws.

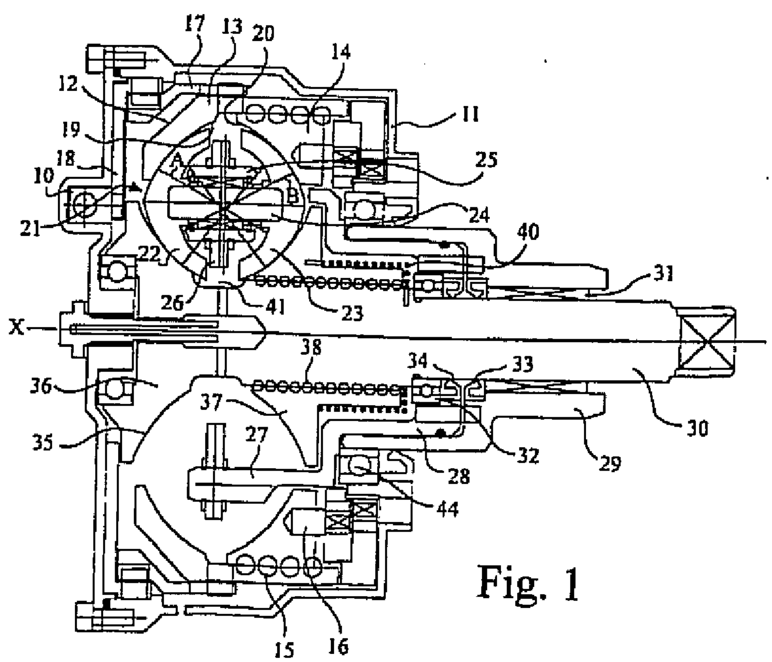

Fig. 1

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At least one drawing originally filed was informal and the print reproduced here is taken from a later filed formal copy.

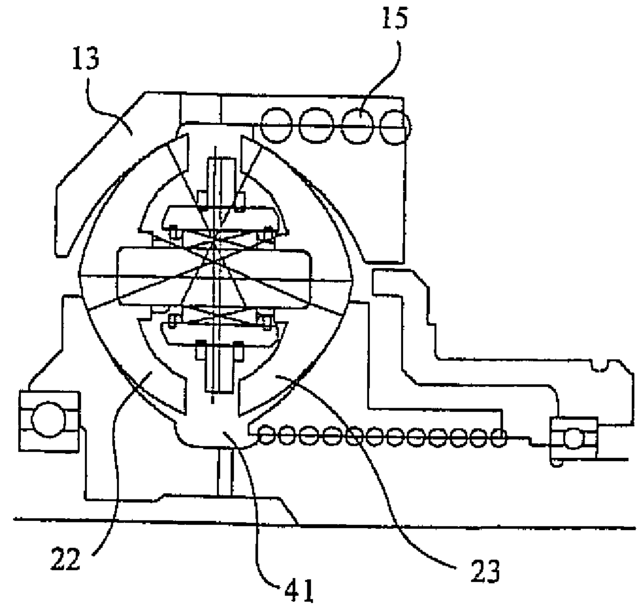
15
13
22
41
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Fig. 3

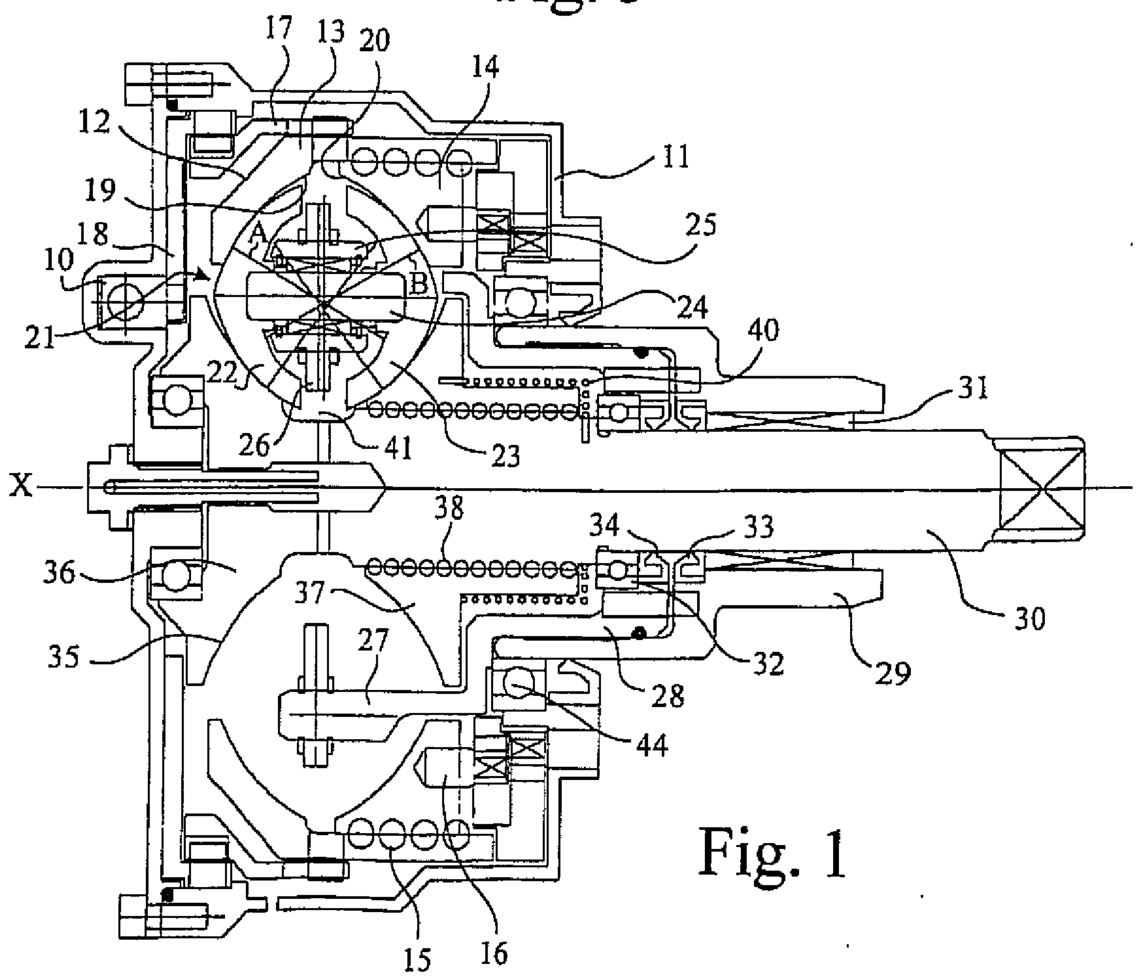
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Fig. 1

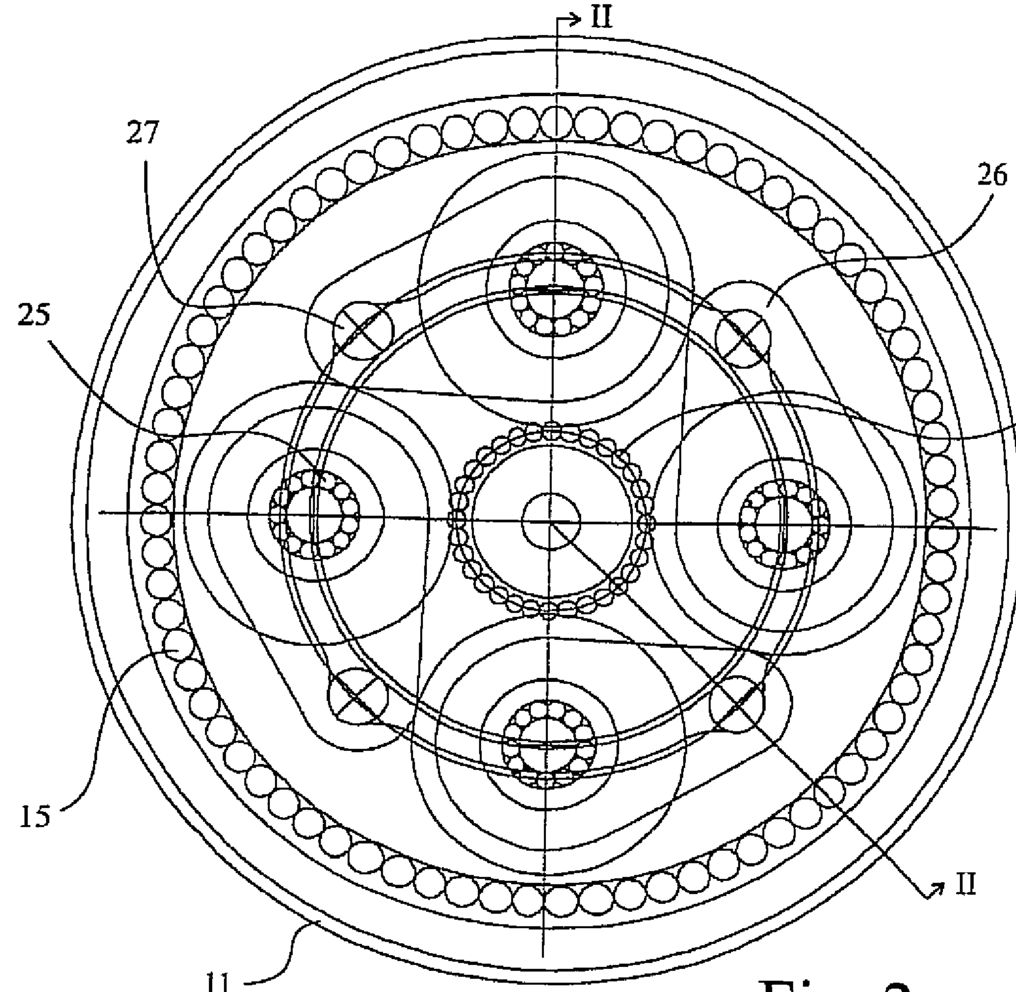


Fig. 2

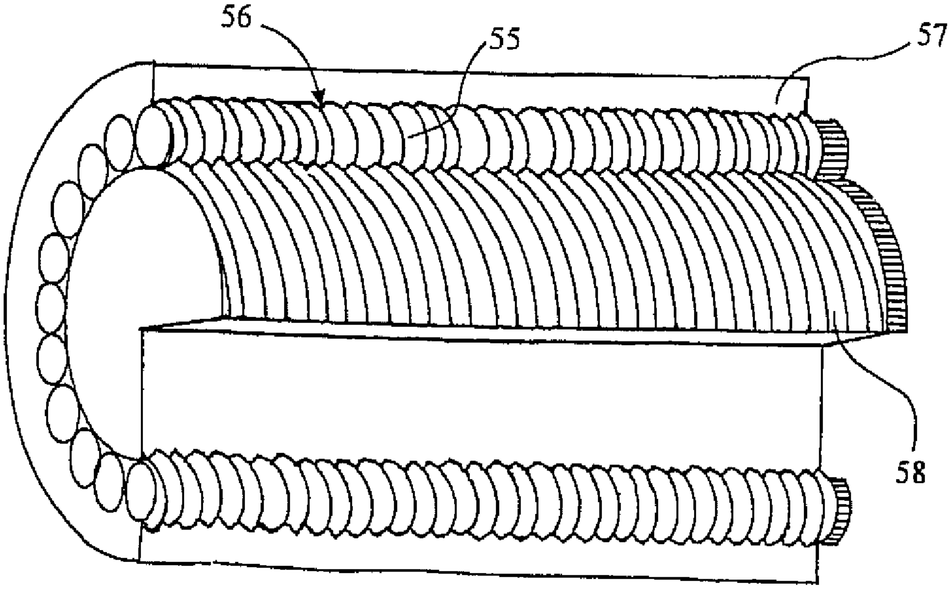


Fig. 6

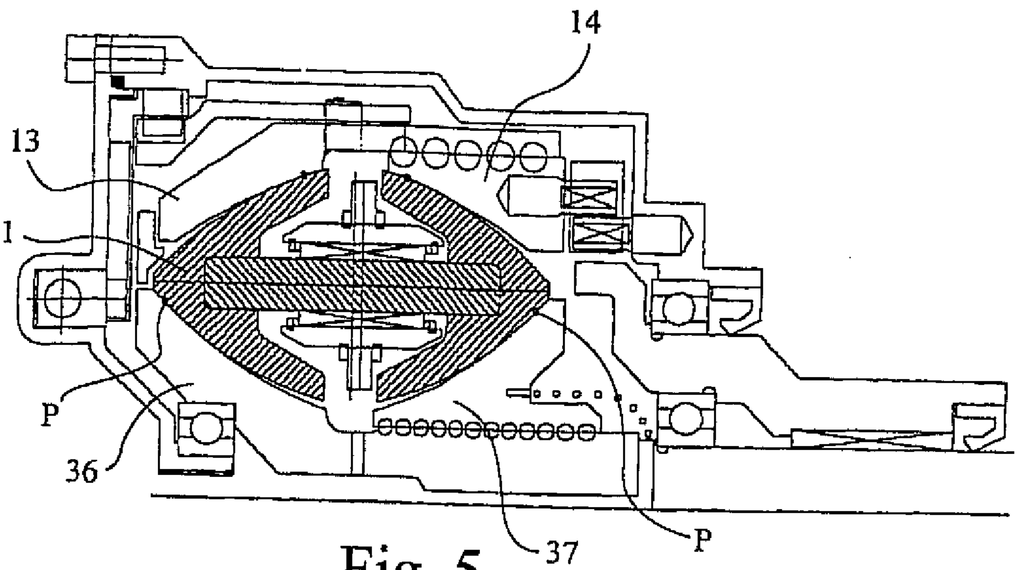
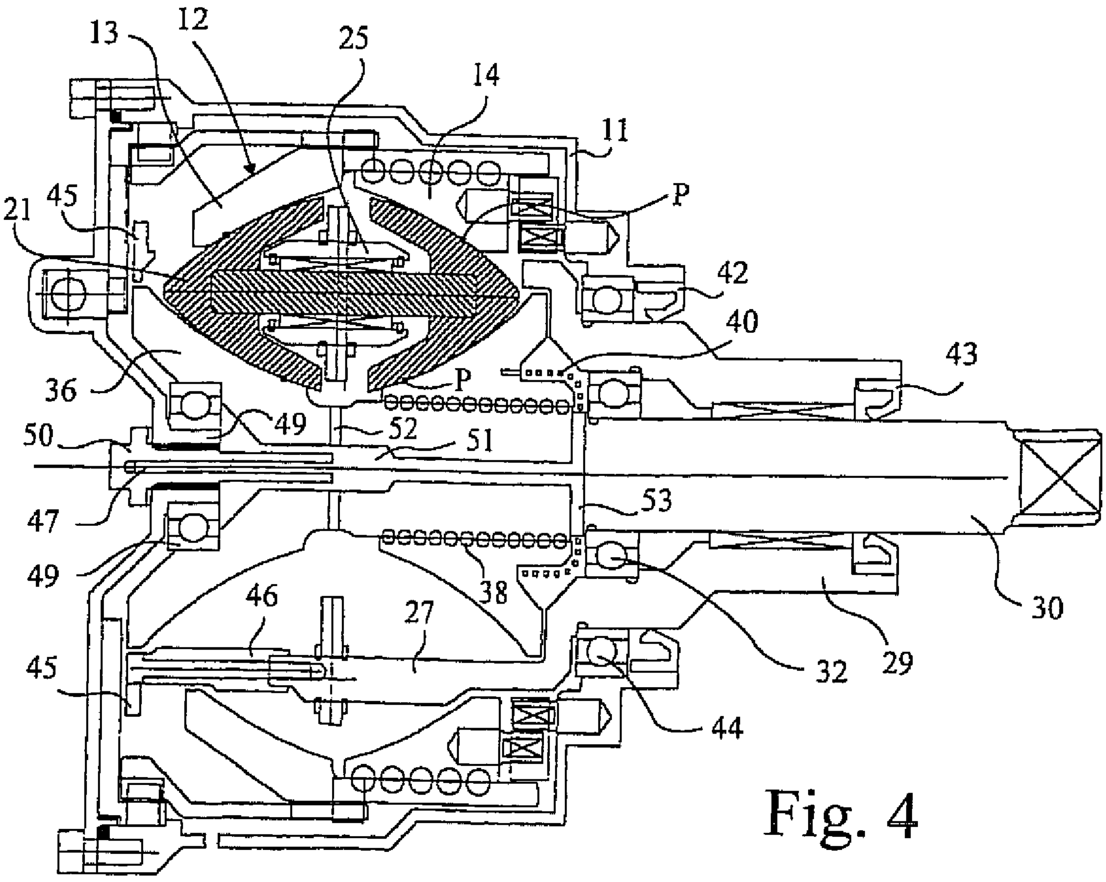

14
13
21
P
36
37
P

Fig. 5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42
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Fig. 4

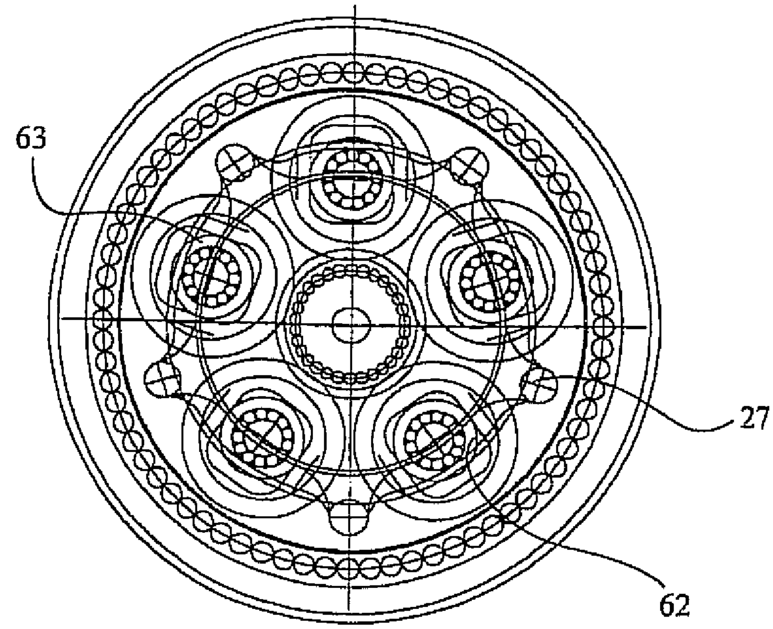


Fig. 8

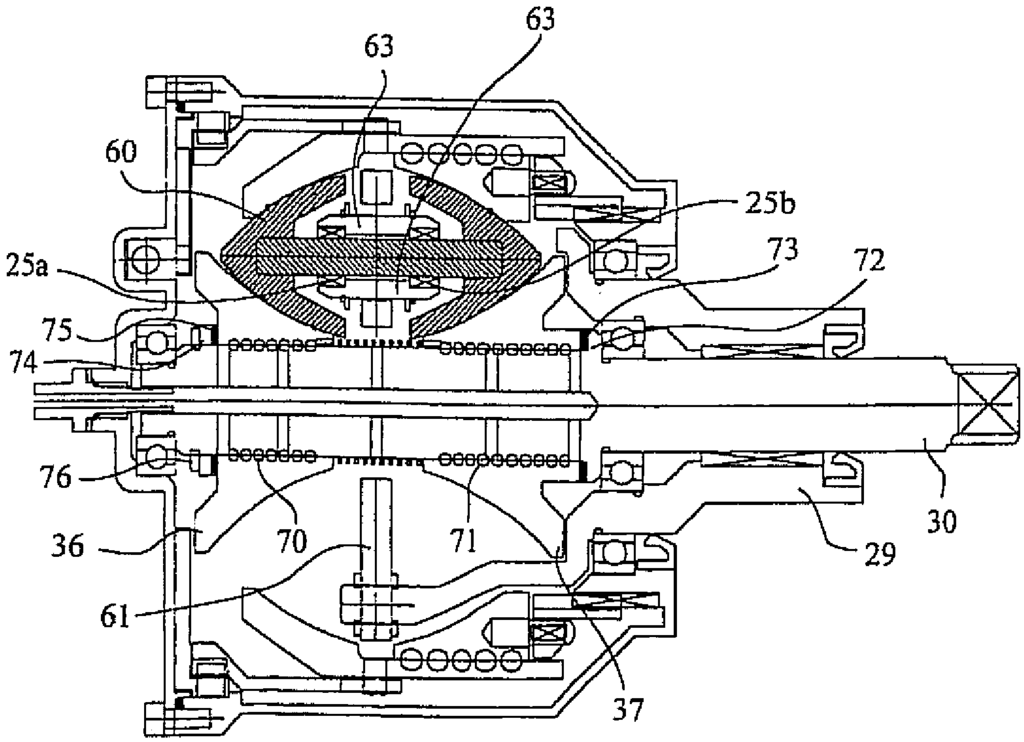


Fig. 7

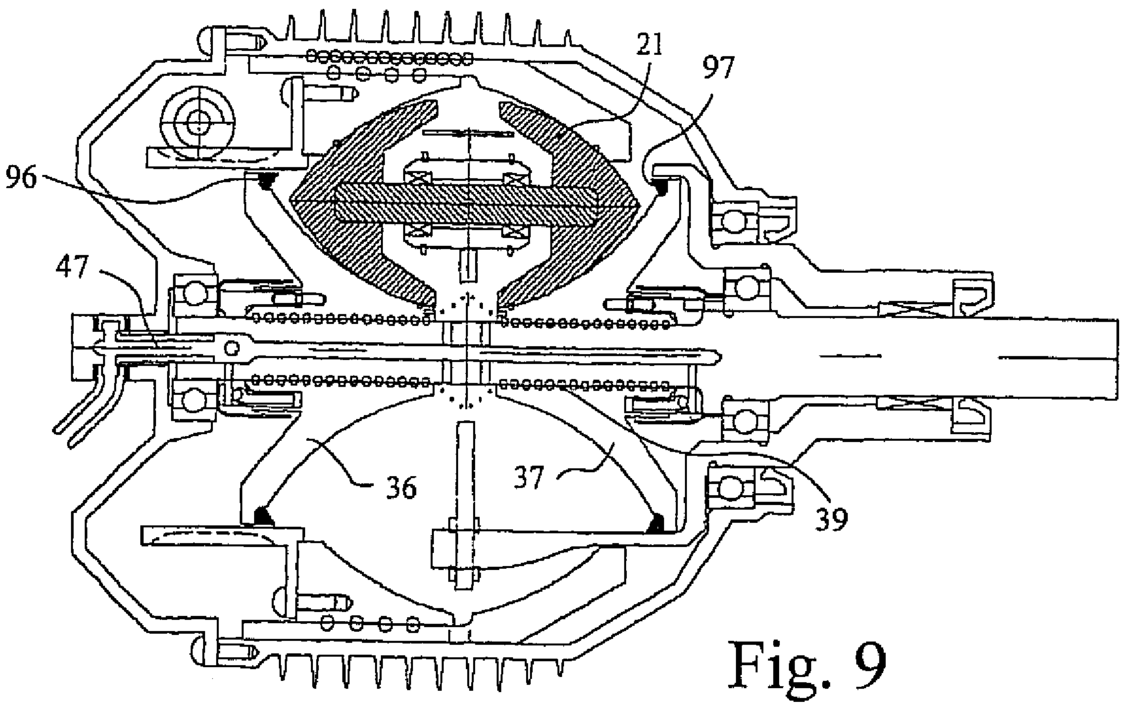


Fig. 9

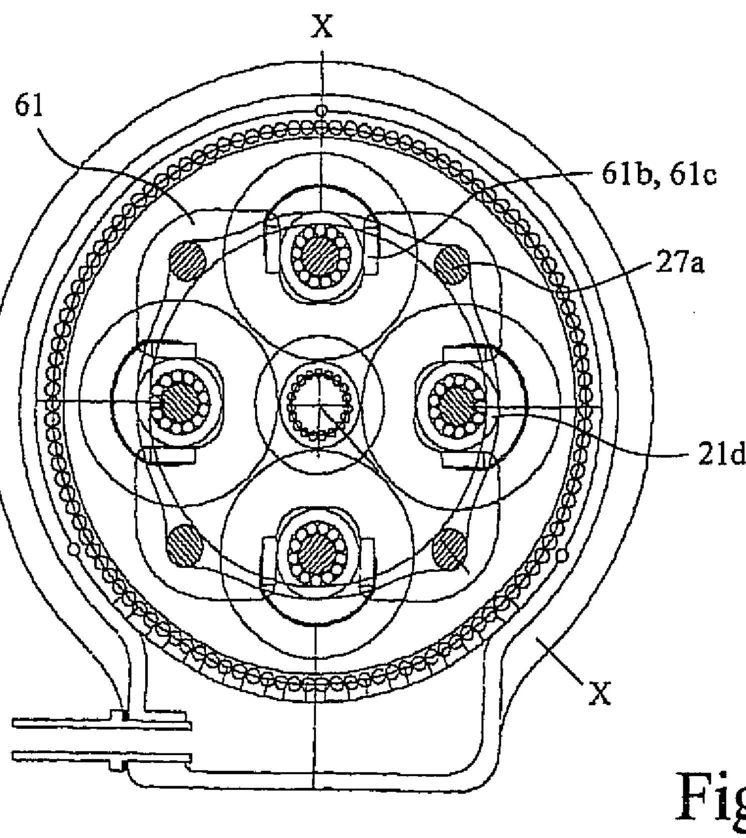
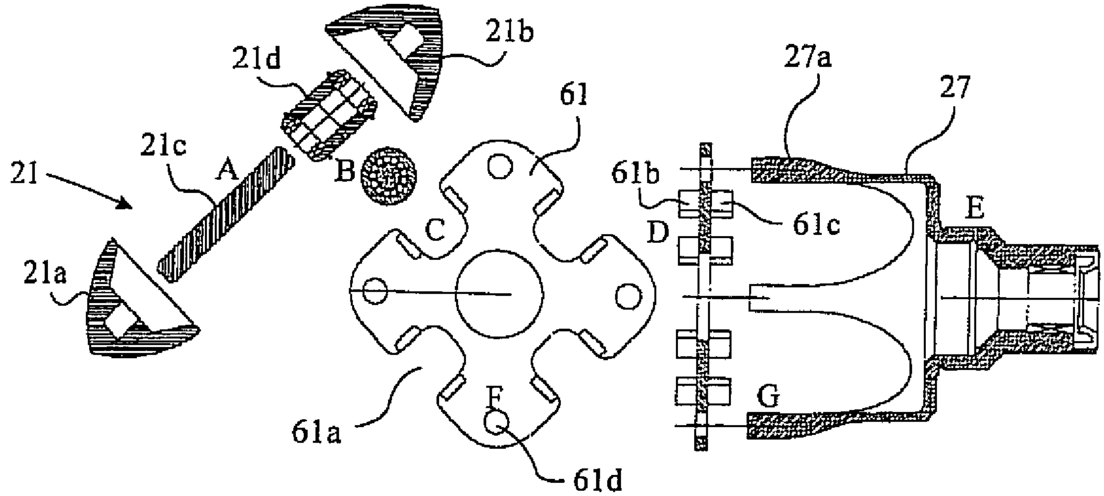


Fig. 10


21b
21d
61
27a
27
21c
A
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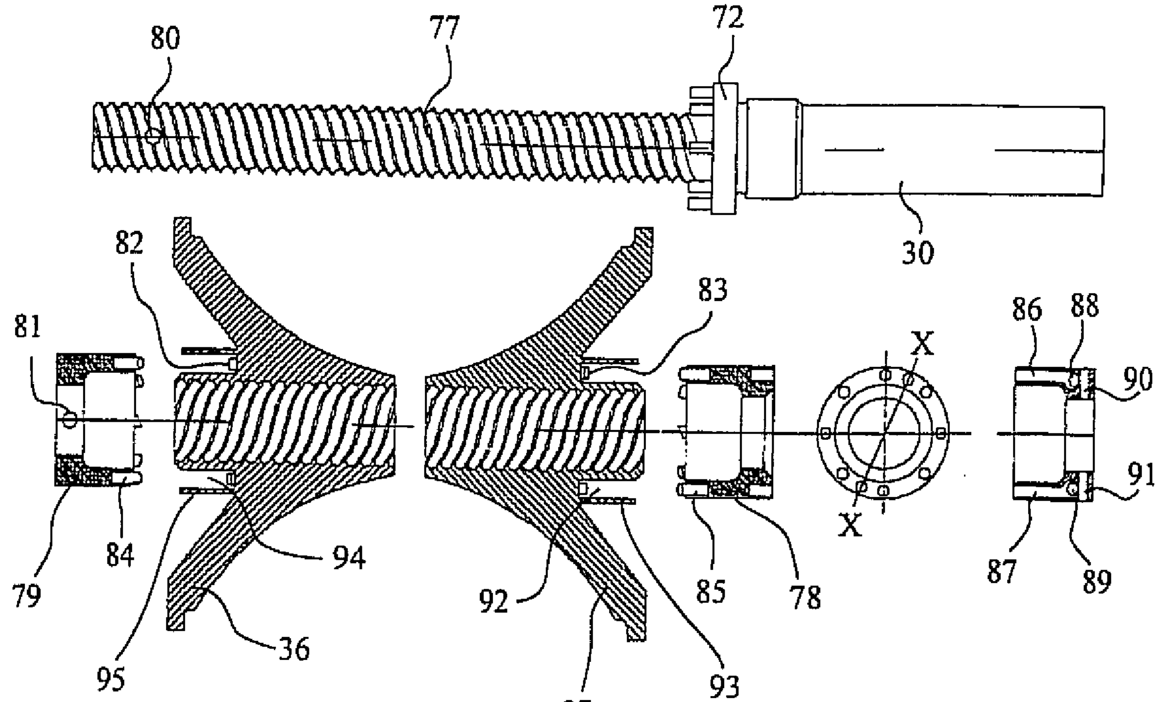
21
61b
61c
E
C
D
21a
F
G
61a
61d

Fig. 11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72
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82
83
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Fig. 12

AN IMPROVED CONTINUOUSLY VARIABLE TRANSMISSION DEVICE

The present invention relates to an improved continuously variable transmission device.

In particular, the present invention relates to a continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith. Such a transmission device may have means sensitive to the torque applied to one of two drive-transmitting members of the transmission (namely the input and output shafts) to determine the compensating variation in the separation of the two parts of the other race and thus the transmission ratio of the device, and also to vary the forces exchanged between the planets and the races normal to the interface between them. The rolling contact between the planetary members and the races is lubricated by means of a very thin film of lubricant. It is essential that this thin film of lubricant be present in order to prevent dry frictional contact between the members in relative motion, which would lead to premature wear, but also that such film should be extremely thin in order to avoid relative slippage.

It is an important design criterion that a transmission device should be most efficient in the transmission ratio most used, that is used for the greatest amount of the time. All transmission devices involve certain losses to friction, and thus heat, and the

design resulting in greatest efficiency is usually applied to the so-called "top" transmission ratio, that is the ratio in which the output shaft rotates fastest for a given speed of rotation of the input shaft. In conventional incremental ratio gearboxes the greatest efficiency is usually achieved when the output shaft is travelling at the same speed as the input shaft to provide a 1:1 or "straight through" transmission ratio. There are, however, circumstances where the transmission ratio at greatest efficiency may be less than 1:1 and, correspondingly situations where a ratio of greater than 1:1 may be desirable.

In a continuously variable rolling contact transmission device of the type defined above the input to the device may be applied via the radially inner races and the output from the device taken from the planets via planet followers or a planet carrier, with the outer race constituting the stationary component. The high gear ratio is then achieved with the two components of the radially outer race located at their position of maximum spacing whilst the parts of the inner race are located as close to one another as possible so that the planets are, effectively, "squeezed" to their greatest radial position. Of course, it will be appreciated that the roles of input and output shaft can be reversed and, in the design in question, the roles of the three components, namely radially inner races, planets assembly, which includes planet followers and planet carriers, and radially outer races are all interchangeable so that any one of them may be held stationary and the other two used either as the input or the output member. It has been found, however, that a configuration as defined above with the outer race stationary has particular advantages from a constructional point of view.

One of the disadvantages arising from this configuration, however, if the planet is a ball, is that in order to obtain the highest ratio possible the patches where rolling contact takes place between the planets and the races are close to their end-of-range positions (closest to the axis of the ball in the case of the radially inner race and furthest from the axis of the ball in the case of the radially outer race). At the end-of-range positions the rolling of the planets over one or the other of the races involves a significant amount of "spin" at the contact patch between the planet and the race, and this generates considerable heat.

The present invention is directed at a rolling contact continuously variable transmission device of the type described above in which the disadvantage of excess heat generation in high transmission ratios is mitigated and a more favourable ratio between the contact patch spin and the roll angular velocity achieved in high ratios.

This is achieved, according to the invention, by changing the shape of the planets from a generally spherical configuration to one involving a spheroidal shape (either a prolate spheroid or an oblate spheroid, which, in essence, allows the contact patch to maintain a more favourable contact angle upon positional variations.

The present invention also involves the direct connection of the planets to the planet carrier by means of a fixed linkage, rather than by way of planet followers which, in earlier arrangements, themselves transferred the forces exerted on the planets to the planet carrier and then to or from the input or output shaft.

According to one aspect of the present invention, therefore, a continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising axially spaced relatively axially movable parts and control means for determining the axial separation of the parts of one of the two races is arranged such that the planets are connected for planetary motion to a planet carrier by a linkage allowing the radial position of the planets to vary in response to variation in the axial separation of the parts of the said one of the two races while maintaining the circumferential connection.

This linkage may be formed as a "trailing link" configuration in which the planet carrier is joined to the planets at an axial mid-point of the planet by providing this latter with a central channel into which the linkage can extend. Transmission of forces is thus symmetrically balanced and a number of other advantages are achieved as will be described in more detail below.

The transmission device of the invention may be put into practice with planets in the form of composite bodies comprising two roller elements each having an outer surface of revolution for engagement with respective parts of the two races. The surface of revolution may be defined by a generatrix which is rectilinear, or has rectilinear, convex or concave portions. Naturally the shape of the races has to correspond to (but not match) that of the planets, being convex towards the planets in the case of rectilinear or concave planetary surfaces and concave in the case of convex planetary surfaces, which latter may be considered the most convenient arrangement.

As discussed above, the planets may be provided with an equatorial channel into which the linkage extends, and in a composite planet member the two individual roller elements of which it may be composed can be joined by an intermediate element to which the said linkage is connected.

The intermediate elements of each planetary body may be joined to the said linkages by roller bearings, preferably needle roller bearings which lie partially within the roller elements. For this purpose each roller element may be shaped as a half-shell. The linkage between each planetary body and the planet carrier may be in the form of a respective trailing arm for each planet. The term "trailing" is of course relevant in only one direction of relative rotation. In the other direction of relative rotation the "trailing" arm becomes a "leading" arm. Forces can be transmitted to and from the planets satisfactorily through such configuration because the planets at the ends of the arms and the linkage to the planet carrier are all constrained to follow a circular motion.

In general the surface of revolution of each roller element of each planetary body is defined by a curvilinear generatrix. This need not be a part of a circle nor, indeed, does it have to be symmetrical or even a regular curve. In one embodiment of the invention the curvilinear generatrix of each roller element surface is, however, an arc of a circle, and preferably the centre of the arc of the circle defining the generatrix for the surface of each roller element is offset axially and/or radially from the midpoint of the planet. If a spherical planet is taken as a standard or exemplary shape, the

preferred shape of the planets for use in the present invention is achieved by effectively displacing the elementary surfaces radially inwardly towards the centre to form a pseudo-spherical body. This is achieved in two ways. If, starting with a notional sphere, an equatorial "slice" were taken through the entire sphere and the two remaining parts were brought together it would have an effect similar to the formation of an oblate spheroid. Similarly, if a cylindrical portion around the axis of rotation of the planetary element extending from one polar region to the other were removed and the remainder of the body effectively compressed in (maintaining the same shape) to replace the material removed, the effect would be to make the surface of the planetary body tend towards a more prolate spheroid. The net result is that the surfaces of the rolling element engaged by the races comprise those parts of a sphere towards the "central" or mid-range of the potential surface of the notional starting sphere, the parts close to the rolling axis and the equatorial region being removed or omitted. This results in a body in which the surface curvature is larger in the rolling direction than in the direction transverse to it. This description of the treatment of a notional sphere, however, is not intended to explain the operations involved in producing a planetary element, but rather only to describe its resultant shape. Being a composite element each of the two axial halves of the rolling element, which are identical to one another, may be manufactured as a "shell" to be held together with their open ends facing one another by an intermediate member as discussed above.

In effect, if the surfaces are generated by generatrices which are in the form of the arc of a circle, the centre of the arc of the circle defining the generatrix for the surface of each roller element is effectively offset axially and/or radially from the midpoint of the planet.

In a preferred embodiment of the invention the planet carrier has a plurality of arms extending from one axial end of the device substantially parallel to the rotation axis of the device, and the free ends of the said arms are reinforced by a reinforcing ring linking together all the said free ends. This reinforcing ring occupies the space between the ends of the arms of the planet carrier and an end cover of the device, lying radially outwardly of the inner races so as not to interfere with the movement thereof.

The said radially inner and outer races are located within a fixed housing and one or other of the said races is rotatable with respect to the housing by the input or output member of the transmission device. In a preferred embodiment of the invention the radially inner race is turnable with respect to the housing with the input member of the transmission. Likewise, it is preferred that the planet carrier is turnable with respect to the housing with the output member of the transmission.

In such a configuration it is possible for the input and output members, which may be, for example, shafts, both to project from the same side of the housing, by forming the output shaft as a hollow member co-axially around the input shaft. This is particularly suitable for use as a transmission for two-wheeled vehicles in which the drive transmission to the driving wheel is effected by means of a chain drive.

In order to ensure lubrication and cooling of the transmission device of the invention various passages are provided for the introduction of a lubricant which also acts as a

coolant as it is pumped through the device. For this purpose one end of the input shaft, preferably that opposite the end projecting from the housing, has a passageway for the introduction of the lubricant axially. This lubricant passageway preferably has a portion extending radially through the input shaft to the region occupied by the said radially inner race parts, and more preferably a median region between the two movable race parts.

In order to achieve relative axial displacement of the two race parts, these may be interconnected by means of a helical coupling, and the frictional inter-engagement can be reduced by the use of rolling elements between the two parts. One of the difficulties encountered with the use of such rolling elements in a helical coupling is the potential for creep of the rolling elements towards one end or the other of the range of movement. If this occurs the rolling element at the end of the track engages a stop preventing it from moving further and increasing frictional contact and reducing the effectiveness of the rolling elements by forcing them to rotate without rolling when engaged against the ends. In order to avoid this difficulty the present invention provides a configuration in which there are provided positive inter-engagement means at each end of the row of rolling elements whereby to prevent relative slip (or creep) between the rolling elements and the race parts in use of the device. Such positive inter-engagement means may, for example, comprise co-operating sets of teeth on the rolling element at one (or each) end of the row thereof and the race parts contacted thereby. This end element can, therefore, only roll upon displacement, with the engagement of the teeth preventing any form of slip.

Alternatively, the rolling elements themselves may have a special conformation. Rather than a spherical or cylindrical element, the rolling element may have a helical surface conformation for engagement with corresponding helical surface formations on the two race parts between which they are located. In effect, the two race parts have co-operating screw threads and the rolling elements, each of which may extend axially for the entire contact distance, has a corresponding screw thread which engages in threads of both the two relatively movable race parts. Any tendency for the rolling element to displace axially as it rolls along the screw thread of one component is countered by the corresponding tendency for it to move axially in the opposite direction by virtue of the rolling displacement along its own screw thread.

One of the limitations on many forms of continuously variable transmission is the inability of the device to transmit torque in both directional senses (to be distinguished from drive transmission in the two directions), in other words, although accelerating drive can be transmitted, decelerating drive, namely when the load drives the driven or output member faster than the drive or input member is driven by the motor cannot. This is familiar to motorists as the engine over-run condition, which allows engine braking of the vehicle. Transmissions which allow only unidirectional torque transmission cannot provide such an over-run facility which, however, is essential for motor transport applications. The transmission of the present invention can provide bi-directional torque transmission. By arranging for the parts of the said other race to be engaged to their associated drive member (whether it be input or output) by a screw coupling of the same hand the two race parts are then urged in the same direction by any torque transmitted through the device, one way or the other,

regardless of the direction of rotation of the drive and driven members. By providing a limit stop at each end of the assembly to limit the movement of the "leading" race part, (and in this context it will be appreciated that for each direction of movement of the race parts with respect to the associated drive or driven members, there will always be a leading and a trailing member, these roles being reversed with a reversal in the direction of relative movement,) then whatever the instantaneous transmission ratio, the two parts of the said other race move together from one end to the other of their associated drive or driven member when a change in torque direction occurs, and the screw coupling maintains the force exerted between them which urges them together. The region of the drive or driven member adjacent the end stops experiences much greater stress (both in torque and in tension) than the rest of the member, because the end stops react only axial forces. In order to provide a large ratio range the radial dimension of the drive or driven member must be kept small. However, larger sections are required to support the greater loads imposed on the system for higher power applications (such as for motor vehicle transmission). For this reason, embodiments of the invention are envisaged in which the end stops are formed with means for reacting torque as well as axial forces. This may be achieved, for example, by forming the end stops as dog clutch stops . A 90^0 dog tooth, which reacts torque but not axial force, would work in theory, but the tooth bearing area would be very small because the mating dog on the race has to approach at a shallow angle as determined by the ball screw helix. The optimum solution lies somewhere between the 90^0 tooth angle of the classic dog tooth and the 0 degree "tooth" angle of the plain stop, and 25^0 is chosen here for best all-round performance in terms of bearing area, shaft stresses and ball screw loads.

The 25^0 contact angle dog drive may be viewed as a second helical engagement mechanism in parallel with the first (the ball screw) but of opposite hand, such that the loads are advantageously shared between the two.

In one embodiment of the invention the said dog clutch arrangement has axially extending teeth with inclined crests the angle of inclination of which is determined by the pitch angle of the flights of the screw thread interengagement between the said two race parts and the said drive transmission member. The present invention may be put into practice with an arrangement in which the said dog clutch arrangement comprises an annular array of axially extending pins or studs on each said part of the said other race and on the said drive transmission member. Preferably the end stop means are carried on respective collars fitted to the said drive transmission means.

The axial oil flow passage within the central shaft of the transmission may also have radial outflow openings to the region at one end of the relatively movable radially inner races to direct coolant lubrication to this region.

Preferably there are provided oil ways through the collars for the passage of lubricating oil, having unidirectional valves opening into an enclosed oil-containing volume defined in part by the respective part of the said other race whereby to provide damping of the motion of the said part of the said other race as it approaches the end stop of the drive transmission member. In a device configured such that the drive transmission member is a central input drive shaft of the device and the said other race

is the radially inner race, the two parts of the said radially inner race act as the cylinders of the damper, with the collars acting as the respective pistons thereof.

The present invention also comprehends a transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which the contact surfaces of the planets are separated into two regions by a median channel in which is engaged a link connecting the planets to a planet carrier.

The present invention also comprehends a transmission device of the general type defined above in which the planets are separated into two parts by a peripheral circumferential groove and the contact surfaces are the surfaces of a body of revolution the generatrices of which are curvilinear lines to form a prolate or oblate spheroid.

Various embodiments of the present invention will now be more particularly described, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is an axial sectional view taken on the line I-I of Figure 2, of a rolling contact continuously variable transmission device formed as an embodiment of the present invention shown in a low transmission ratio;

Figure 2 is an end view of the embodiment of Figure 1;

Figure 3 is an axial sectional view of a part of the embodiment of Figure 1, illustrating the components in a high transmission ratio;

Figure 4 is an axial sectional view similar to that of Figure 1, illustrating a second embodiment of the invention in which the planetary members are prolate spheroids shown in a configuration for a low transmission ratio;

Figure 5 is a similar axial view of part of the embodiment of Figure 4 showing the movable components in a configuration for a high transmission ratio;

Figure 6 is a schematic view of a part of a further embodiment of the invention showing a modified helical interengagement arrangement;

Figure 7 is an axial sectional view of a further embodiment of invention:

Figure 8 is an end view of the embodiment of Figure 7;

Figure 9 is an axial sectional view of a further embodiment of the invention designed both for heavy loads and a large ratio range;

Figure 10 is an end view of the embodiment of Figure 9;

Figure 11 is an exploded view of the power take-off components of the embodiment of Figures 9 and 10; and

Figure 12 is an exploded sectional view of the inner race component set of the embodiment of Figures 9 and 10.

Referring now to the drawings, and particularly to Figures 1, 2 and 3, the transmission device shown comprises a housing generally indicated 11 within which is located a radially outer race 12 formed in two relatively axially displaceable parts 13, 14 engaged together by a so-called "ball screw" comprising several helical rows of balls 15 engaged in corresponding helical grooves in the two parts 13, 14 allowing them to

turn with respect to one another about the central longitudinal axis X-X of the device. The ball screw has several starts (four in this case); this results from the need to fill the space available with balls (for maximum load capacity) but to avoid using large balls (which would be required for a single start thread) with the relatively long lead needed to balance the axial and circumferential loads. Relative axial displacement between the two parts 13, 14 is achieved by mounting the part 14 on fixed pins 16 which form an Oldham coupling with a pair of pins in the housing to restrain the race part 14 against rotational motion whilst allowing axial displacement. The Oldham coupling is used here as a "tolerance accommodating" arrangement which allows radial translation but not rotation. The two pairs of pins do not in fact lie in the same plane, as appears to be shown in the drawing, but are disposed at 90° from each other and the small flats indicated by the crosses run in slots in the Oldham ring. The rotationally displaceable race part 13 is held in a generally cylindrical holder 17 which can be turned about the axis X-X by an adjuster arm 18 turned by an adjuster actuator 10. The actuator 10, seen end-on in Figure 1 is preferably a screw actuator having a ball screw driven by an electric motor (not shown). By turning the rotationally displaceable race part 13 about the axis X-X this is itself effectively "screwed" in relation to the axially displaceable outer race part 14 by the action of the ball screw 15, causing this to move axially along the slide pins 16 without turning. In this way the two race parts 13, 14 are moved apart or together by turning the rotationally displaceable outer race part 13 in one direction or the other. The two race parts have curved race surfaces 19, 20 engaged by the curved surfaces of a planetary member generally indicated 21 which comprises two approximately hemispherical shells 22, 23 held together by a central pin 24 which carries a rolling element bearing 25 by

which the planet member 21 is carried on a respective connecting link 26. As can be seen in Figure 2 each connecting link 26 is connected to a planet carrier arm 27 of a planet carrier 28 which is fixedly connected to an output shaft 29 which co-axially surrounds and is borne on the input shaft 30 by means of a bearing 31. A further bearing 32 interconnects the input shaft 30 and the planet carrier 28, and seals 33, 34 protect the interior of the device from ingress of dust, dirt and other contaminating particles, humidity or moisture.

The planet members 21 also roll on an inner race generally indicated 35 comprising an axially fixed race part 36 and an axially displaceable race part 37 carried thereon by a ball screw 38 similar to that by which the two parts of the radially outer race are interconnected. A light pre-loading torsion spring 40 urges the axially displaceable inner race part 37 towards the planet member 21 in order to maintain contact.

The manner in which transmission ratios are changed and the torque between the input and output shaft is sensed by the axially displaceable radially inner race part 37 carried by the ball screw 38 on the axially fixed race part 36 is described in our above-mentioned International Patent Application No. WO99/35417, the disclosure of which is incorporated herein by reference and will not be further described here except in relation to the shape of the planet members 21.

In the earlier International Application referred to above the planets were spherical solid balls and the forces exerted by their motion between the radially inner and outer races were transmitted via planet followers located between each adjacent pair of

planets. When the outer race parts are moved together in order to urge the planets radially inwardly the radially inner race parts were forced apart with the contact pressure being maintained by the torque-sensitive configuration as explained in that document. As the two radially outer race parts approach their position of closest approach the contact patches between the planets and the races move radially inwardly and, by virtue of the shape of the spherical planets, the normal to the contacting surfaces, which passes through the centre of the planet, becomes more shallowly inclined with respect to the rolling axis so that the radially resolved component of force becomes smaller and the axially resolved component greater. A very much larger absolute contact force on the planet must, therefore, be exerted in order to reach the lower ratios and, of course, there comes a point at which the additional radial displacement available by further increasing the force becomes relatively small and the forces become unacceptably high. Moreover, in the highest and lowest ratio the contact patches closest to the rolling axis of the planet experience substantial "spin" increasing the heating effect of the frictional contact thereby generating additional heat which needs to be dissipated in order to maintain the device within tolerable limits. By contrast to the shape of a spherical planet, however, the configuration of the planet member 21 in the present invention exploits only those sectors of the circumferential generatrix which are most effective, with the equatorial band being omitted due to the presence of the annular space 41 between the two shells 22, 23 and the shape of the polar regions being modified by the shape of each of the shells 22, 23. In this embodiment, as can be seen from Figure 1, in the lowest ratio achievable, where the outer race parts 13, 14 contact the planet member 21 closest to its rolling axis the contact surfaces are still inclined at an angle in the region of 30°

and, moreover, the intersection of the normals to the contact surfaces, represented by the lines A and B in Figure 1 intersect at a point offset from the centre of the spheroid defined by the curved running surfaces of the planet member. This limits the spin of the contact patches and enables the device to bear greater loads. The direct connection of the planet members 21 via links 26 to the planet carrier arms 27 also enables the device to sustain a greater load by allowing more planets to be fitted into the available space.

Referring now to Figures 4 and 5 there is shown an alternative embodiment in which the planetary members have a more pronounced prolate shape. In the embodiment of Figures 4 and 5 those components which correspond to or fulfil the same function as corresponding components in the embodiments of Figures 1 to 3 have been identified with the same reference numerals. In this embodiment, however, the arms 27 of the planet carrier are formed integrally with the output shaft 29 so that the bearing 32 between the input shaft and the planet carrier now acts directly between the input and the output shaft, and the bearing 44 between the output shaft and the casing, previously axially offset from the bearing 32, is now in close axial alignment therewith, increasing the strength of the device. The seals 33, 34 are replaced by a seal 43 between the end of the output shaft and the input shaft 30, and a seal 42 protects the bearing 44 from the ingress of dust, dirt and other contaminants between the output shaft 29 and the casing 11.

The arms 27 of the planet carrier have extension pieces 46 secured thereto carrying a reinforcing ring 45 in a position immediately surrounding the end of the axially

fixed inner race part 36. This inner race part 36 is borne by a bearing 48 on a boss 49 of the casing 11, within which is located a central plug 50 having an axially extending passage 47 for the introduction of the cooling lubricant into the interior of the device. The passage 47 in the plug 50 opens into a chamber 51 within the combined input shaft 50 and inner race 35 from which extend two radial passages 52, 53 the first being axially located in register with the space 41 between the shells 22, 23, allowing cooling lubricant to be injected directly into contact with the bearings 25 between the planet members 21 and the links 26, and the second (53) opening into the region of the main bearing 32 between the output shaft 29 and the input shaft 30. The chamber 51 also extends to the bearing 49 so that oil injected into the central passage 47 in the shaft 30 can be directly applied to the main bearings 49, 32, the radially inner ball screw 40 and the bearings 25 of the planet members 21. The additional cooling and lubrication insured by this force flow of lubricant, together with the presence of the reinforcing ring 45 and the prolate shape of the planet members 21 enables a higher load-bearing capacity to be achieved.

As can be seen in Figure 4 the highly prolate spheroid shape of the planet members 21 ensures that the normal to the contact patches, illustrated by the dots P in Figure 4, remain inclined at more than 45° to the rolling axis of the planet member 21 even upon closest approach to this axis. The resolution of forces into radial and axial components can then be seen to favour the radial component even when the two parts 13, 14 of the radially outer race 12 are in their position of closest approach (Figure 4) without detrimentally affecting the axial component of the forces exchanged between the planet members 21 and the parts 36, 37 of the radially inner race 35. Likewise,

as can be seen in Figure 5, where the radially outer race parts 13, 14 are shown in their position of greatest separation, the angles between the normals to the contact patches P between the planet member 21 and the radially inner race parts 36, 37 remains inclined in the region of 45° to the rolling axis of the planet thereby favourably increasing the radial component of this force in comparison with the radial component of the corresponding force in a spherical planet.

It will be appreciated that arrangements to allow bi-directional torque transmission through a device such as the transmission device described above can be introduced using the ideas explained in co-pending application number 0016261.0 the disclosure of which is incorporated herein by reference.

One of the problems associated with ball screws such as the screws 15, 38 used to interconnect the two parts of the radially inner and radially outer races 12, 35 lies in the fact that slip or "creep" between the balls and the raceways in which they are housed can result in the balls at the ends of the row engaging end stops and being prevented from executing their normal rolling motion. This can be countered by providing the balls at each opposite end of the row with teeth meshing in corresponding teeth or serrations at the end regions of the helical channels. This does not compromise the load bearing capability of the remainder of the ball screw whilst allowing certainty in the rolling action ensuring that no slip between the balls and the channels takes place.

In the alternative configuration illustrated in Figure 6 the balls may be replaced by

rollers 55 having helical grooves 56 which engage in corresponding helical grooves 57, 58 in the radially outer and radially inner components between which the rollers 55 are located. Naturally the pitch and number of threads in each component is the same and preferably the rollers have a single start thread which may preferably be a triangular thread with an included angle of 90° although the thread form may be barrelled in order to ensure a large contact radius. Because the threads all have the same pitch the rollers are not shifted axially as they roll between the two members, any tendency to move axially in one direction by the thread on one of the members being countered by the tendency to move axially in the opposite direction by the thread upon the roller. At each end the rollers have gear teeth which mesh with toothed rings on the two members between which the rollers are engaged in order to ensure correct rolling motion without any slip.

Referring now to Figures 7 and 8, the alternative embodiment shown is configured to allow maximum use to be made of the circumferential space so that the greatest possible number of planets can be fitted in a device of a given size. In Figures 7 and 8, as in the previous embodiments, the same reference numerals are used to indicate the same or corresponding components. As will be seen from Figure 8 this embodiment has five planets 60 in a transmission of the same dimensions as the embodiment of Figure 1 which has only four planets. These planets 60 are linked to the arms 27 of the carrier by a disc 61 fixed to the arms 27 of the planet carrier in the median plane of the ring of planets 60. The disc 61 has wide generally radial slots 62 within which are housed bushes 63 which house rolling element bearings 25a, 25b on which the planets roll. The bushes 63 themselves roll within the slots 62 during ratio

changing movements. The slots could be inclined from the strictly radial orientation, and this allows the contact forces at the inner race to be increased or reduced while those at the outer race are reduced or increased respectively. This can be a useful design tool.

This embodiment is circumferentially very compact and has a high load-bearing capacity. The disc 61 is thickened locally to provide wider support for the rollers constituted by the bushes 63 and, of course, it is not necessary to extend the arms 27 of the planet carrier to a reinforcing disc as in the embodiment of Figure 4 since the disc 61 itself provides a much greater stiffness. This embodiment also allows bi-directional torque transmission between the input shaft 30 and the output shaft 29. For this purpose the ball screw 38 of, for example, Figure 1, between the input shaft 30 and the right hand inner race half 37 is replaced by a coaxial ball screw coupling 70, 71, in the form of respective rows of balls in co-operating thread flights in the input shaft 30 and both the left and right inner race halves 36, 37. Both ball screws are the same hand so that a given direction of torque transmission will cause both inner race halves to tend to be driven axially along the input shaft 30 in the same direction, for example to the left for positive drive torque transmission and to the right for over run or negative drive torque transmission.

The drive shaft 30 has a central flange 72 forming a shoulder with an annular wear pad 73, and the left hand end of the drive shaft 30 carries an annular abutment end stop 74 with a corresponding annular wear pad 75. The member 74 is retained in position by a circlip 76 engaged in an annular groove in the end of the drive shaft 30.

The two end stops 72, 74 engage corresponding radial surfaces of the right and left inner race halves 37, 36, respectively. Thus, during positive drive transmission when the two race halves 36, 37 are both driven to the right as viewed in the drawing, the abutment 72 limits the movement of the race half 37 so that the continued screwing action on the race half 36 maintains the squeezing force on the planets 60. Correspondingly, for negative torque transmission, the two race halves 36, 37 travel to the left on their respective ball screws 70,71 until the race half 36 engages the end stop 74 and the continued screwing action of the ball screw 70 drives the race half 37 towards the race half 36, again maintaining the squeezing action on the planets 60. Depending on the precise transmission ratio in force at the time of torque reversal there may be a more or less significant shift in the position of the race halves 36, 37. That is, in a low ratio, when the race halves 36, 37 are at their maximum separation, there may even be no (or at least very limited) displacement along the ball screws 70,71. On the other hand, in the highest ratio when the radially inner race halves 36, 37 are at their position of closest approach there is the maximum separation between their opposite radial faces and the end stops 72, 74 so that a maximum axial displacement takes place. This can lead to a noticeable impact of the race half against the corresponding end stop upon torque direction change, and the embodiment of Figure 9 is provided with means for overcoming any disadvantages associated with this.

The embodiment of Figure 9 is also configured, as mentioned above, for supporting a heavier load and obtaining a large ratio range as will be explained in more detail below. Again, as with the earlier embodiments, those components which are the same

as or fulfil corresponding functions to the embodiments described earlier will be identified with the same reference numerals. In this embodiment the inner race component set, particularly illustrated in Figure 12, comprise the two inner race halves 36, 37 mounted by a ball screw configuration on a threaded part 77 of the input shaft 30 by balls 39 (see Figure 9) which are not shown in Figure 12. In place of the end stops 72, 74 the embodiment of Figure 9 has two collars 78, 79 the former of which is engaged on the flange 72 of the d rive shaft 30 and the latter of which is secured to the opposite end of the drive shaft 30 by means of a shear pin (not shown) passing through aligned holes 80 (in the shaft 30) and 81 (in the collar 79).

In order to permit a large ratio range it is necessary for the ballscrew section of the shaft 30 to have as small a diameter as possible. For heavier loads, however, the shaft requires to be more robust. In order to share the load a dog-clutch arrangement between the inner race halves 36, 37 and the collars 79, 78 is provided. This comprises an annular array of axially extending pins 82, 83 on the race halves 36, 37 and an annular array of pins 84, 85 on the collars 79, 78. The end faces of these pins are inclined to allow them at least partially to react both torsional and axial loads, bearing in mind that the engagement of the dog clutch thus formed occurs with a relatively helical motion of the raceway parts 36, 37 on the screw threaded part 77 of the shaft 30.

For damping the backlash which occurs upon torque reversal as discussed in relation to Figure 7, the embodiment of Figures 9 to 12 is provided with a pair of axial passageways 86, 87 controlled by respective unidirectional valves 88, 89 which

allows oil in the central channel 47 to pass radially, via respective radial passages 90, 91, through the unidirectional valves 88,89 into the axial passages 86,-87 and from there into an annular chamber 92 defined between the race way half 37 and the collar 78 by an annular tubular sleeve 93. The left hand raceway half 36 has a similar annular chamber 94 defined by sleeve 95 for oil entering through passages (not shown) in the collar 79. Oil under pressure in these chambers 92, 94 can only escape through the small gap between the collar acting as the (annular) piston and the cylinder defined by the race way half as the race is driven towards the stop on torque reversal. This damps the backlash and prevents impact noise as metal-to-metal drive is re-established.

A typical planet 21 is illustrated as part of the power take-off system shown in Figure 11, and comprises 2 planet halves 21a, 21b which are a push fit on a central shaft 21c which is first passed through the sleeve 21d having appropriate cylindrical needle bearings at each end. The sleeve 21d engages in a radial slot 61a in the plate 61 (in this embodiment an approximately square plate with rounded corners) which slot has axially extended sides 61b, 61c to provide a widened surface on which the sleeve is able to roll a small distance in a radial direction. The plate 61 delivers its power to the carrier 27 via holes 61d into which fingers 27a of the carrier engage.

In all of the embodiments described above it is possible to arrange for a top gear transmission ratio in which the contact patch spin is avoided by limiting the maximum possible radially outward excursions of the planets to be less than that allowed by the radially outer race when the two parts thereof are separated beyond a threshold value.

This can be achieved in the embodiments of Figures 1 to 6 by limiting the rotation of the links 26, for example by means of an abutment stop (not shown) mounted on the carrier to arrest outward excursion of the link 26. This could be positioned at some point between the end connected to the planet and the end connected to the planet carrier. In the embodiment of Figures 7 and 8 this objective could be achieved by ensuring that the radially outer ends of slots 62 hold the planets to a radially outward excursion less than that allowed by the maximum separation of the parts of the radially outer race. In the embodiment of Figures 9 to 12, this top gear lock up is provided by forming the inner race halves 36, 37 with respective grip rings 96, 97 at their periphery, and shaping them such that the grip rings engage the planets 21 on the radially outer side of the rolling axis. Thus, when the radially outer races 13, 14 are separated to their maximum extend, allowing the radially inner races to reach their position of closest approach driven by the ball screw 39, the two grip rings 96, 97 engage the planets 21 to form a direct drive between the input shaft and the output shaft. In this configuration, which requires there to be clearance between the planet 21 and the outer races 13, 14, there is a, step change, typically in the region of 1.2: 1, between the highest rolling ratio and this locked top gear since the planets 21 are held against rotation by the radially inner race halves in this condition.

CLAIMS

1. A continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising axially spaced relatively axially movable parts, and control means for determining the axial separation of the parts of one of the two races, in which the planets are connected for planetary motion to a planet carrier by connection means which allows the radial position of the planets to vary in response to variation in the axial separation of the parts of the said one of the two races while maintaining the circumferential connection.

2. A continuously variable transmission device according to Claim 1, in which the planets are composite bodies comprising two roller elements each having an outer surface of revolution for engagement with respective parts of the two races.

3. A continuously variable transmission device according to Claim 2, in which the two roller elements are joined by an intermediate element to which the said connection means is connected.

4. A continuously variable transmission device according to Claim 3, in which the intermediate elements of each planetary body are joined to the said connection means by roller bearings, preferably needle roller bearings.

5. A continuously variable transmission device according to any preceding claim,

in which the connection means between the planets and the planet carrier comprises a respective trailing arm for each planet.

6. A continuously variable transmission device according to any preceding claim, in which the connection means between the planets and the planet carrier comprises a connector plate having a plurality of slots, having at least a radial component, within each of which a part of a respective planet is engaged.

7. A continuously variable transmission device according to Claim 6, in which the slots of the connector plate engage respective bushes of the planets within which are located rolling element bearings on which the planets rotate.

8. A continuously variable transmission according to any preceding claim, in which the surface of revolution of each roller element of each planet is defined by a curvilinear generatrix.

9. A continuously variable transmission device according to Claim 8, in which the curvilinear generatrix of each roller element is an arc of a circle.

10. A continuously variable transmission device according to Claim 9, in which the centre of the arc of the circle defining the generatrix for the surface of each roller element is offset axially and/or radially from the mid point of the planet.

11. A continuously variable transmission device according to any preceding claim,

in which the planet carrier has a plurality of linkage support arms extending substantially parallel to the rotation axis of the device from one axial end of the device, and the free ends of the said linkage support arms, are reinforced by a reinforcing ring linking together all the said free ends.

12. A continuously variable transmission device according to any preceding claim, in which the said radially inner and outer races are located within a fixed housing and one or other of the said races is rotatable with respect to the housing by the input or output shaft of the transmission device.

13. A continuously variable transmission device according to Claim 12, in which the radially inner race is turnable with respect to the housing with input shaft of the transmission.

14. A continuously variable transmission device according to Claim 13, in which the planet carrier is turnable with respect to the housing with the output shaft of the transmission.

15. A continuously variable transmission device according to any of Claims 11 to 13, in which the output shaft extends coaxially around the input shaft and both project from the same side of the housing.

16. A continuously variable transmission device according to Claim 15, in which one end of the input shaft opposite the end projecting from the housing has a

passageway for the introduction of lubricant.

17. A continuously variable transmission device according to Claim 16, in which the lubricant passage has a portion extending radially through the input shaft to the region occupied by the said radially inner race parts.

18. A continuously variable transmission device according to any preceding claim, in which the planet carrier is formed in one piece with output or the input shaft.

19. A continuously variable transmission device according to any preceding claim in which the two parts of the radially outer race and/or the radially inner race are interconnected by means of a helical coupling, with rolling elements between the two parts to reduce friction.

20. A continuously variable transmission device according to Claim 19, in which there are provided positive interengagement means at each end of the row of rolling elements whereby to prevent relative slip between the rolling elements and the race parts in use of the device.

21. A continuously variable transmission device according to Claim 20, in which the positive interengagement means comprise co-operating sets of teeth on the rolling element at the (or each) end of the row thereof and the race parts contacted thereby.

22. A continuously variable transmission device according to Claim 19, in which

the rolling elements themselves have a helical surface conformation for engagement with corresponding helical surface formations or the two race parts between which they are located.

23. A continuously variable transmission device according to any preceding claim, in which the other of the two races is interengaged with an associated drive transmission member (input or out put; drive or driven) by a screw coupling of the same hand, there being respective end stop means for limiting the travel of the associated race part of the said other race in a respective directional sense whereby to permit torque transmission through the device in both senses.

24. A continuously variable transmission device according to Claim 23, characterised in that the said end stop means includes means for torsionally interengaging the drive transmission member with the said respective race part in the end-of-travel position thereof.

25. A continuously variable transmission device according to Claim 23, characterised in that the said means for torsionally interengaging the said drive transmission member comprises a dog clutch arrangement.

26. A continuously variable transmission device according to Claim 25, characterised in that the said dog clutch arrangement has axially extending teeth with inclined crests the angle of inclination of which is determined by deference to the pitch angle of the flights of the screw thread interengagement between the said two

race parts and the said drive transmission member.

27. A continuously variable transmission device according to Claim 25 or Claim 26, characterised in that the said dog clutch arrangement comprises an annular array of axially extending pins or studs on each said part of the said other race and on the said drive transmission member.

28. A continuously variable transmission device according to any of Claims 23 to 27, characterised in that the end stop means are carried on respective collars fitted to the said drive transmission means.

29. A continuously variable transmission device according to Claim 28, characterised in that there are provided oil ways through the collars for the passage of lubricating oil, having unidirectional valves opening into an enclosed oil-containing volume defined in part by the respective part of the said other race whereby to provide damping of the motion of the said part of the said other race as it approaches the end stop of the drive transmission member.

30. A continuously variable transmission device according to Claim 29 characterised in that the drive transmission member is a central input drive shaft of the device and the said other race is the radially inner race, the two parts of the said radially inner race acting as the cylinder of the damper, with the collars acting as the respective pistons thereof.

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Application No:	GB 0220741.3	**Examiner:**	Mike Mckinney
Claims searched:	1 to 30	**Date of search:**	4 March 2003

Patents Act 1977 : Search Report under Section 17

Documents considered to be relevant:

Category	Relevant to claims	Identity of document and passage or figure of particular relevance	
A		GB 2354293 A	(MILNER) = GB0016261.0, referred to on page 19 of the description.
A		WO 99/35417 A1	(MILNER) referred to on page 15 of the description.
A		US 5390558	(WEINBERG)
A		US 5318486	(LUTZ)
A		US 3516305	(CHERY)

Categories:

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
Y	Document indicating lack of inventive step if combined with one or more other documents of same category.	P	Document published on or after the declared priority date but before the filing date of this invention.
&	Member of the same patent family	E	Patent document published on or after, but with priority date earlier than, the filing date of this application.

Field of Search:

Search of GB, EP, WO & US patent documents classified in the following areas of the UKC^{V}:

F2D

Worldwide search of patent documents classified in the following areas of the IPC^{7}:

F16H

The following online and other databases have been used in the preparation of this search report:

EPODOC, JAPIO, WPI

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ONLINE: WPI; EPODOC; JAPIO.

(54) Abstract Title

Planetary-ball continuous variable transmission

(57) A planetary-ball continuous variable transmission comprises planetary members, eg planetary balls 15, in rolling contact with radially inner and outer races 14, 13 each having axially spaced parts 14a, 14b, 13a, 13b. Control means comprising a screw 29 operated by a Bowden cable (30, fig 5) determines axial displacement of the outer races 13a, 13b and thereby a radial position of the balls 15. Axial displacement of the outer races 13a, 13b causes, via balls 15, axial displacement of the inner races 14a, 14b and the forces exerted on the balls 15 is regulated by a bi-directional torque-sensitive coupling comprising a screw thread on shaft 11 engaging in screw thread in inner races 14a, 14b. The planetary balls 15 may have peripheral grooves (25, fig 2) which receive planet followers (16) and contact surfaces of the planetary balls 15 may have a plurality of conical surfaces (26, fig 6) to form annular facets which define a given gear ratio.

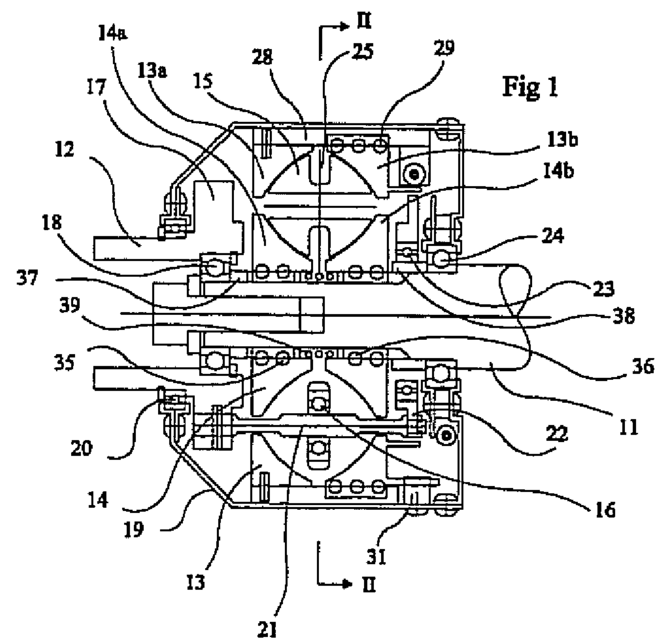


Fig 1

GB 2 342 130 A

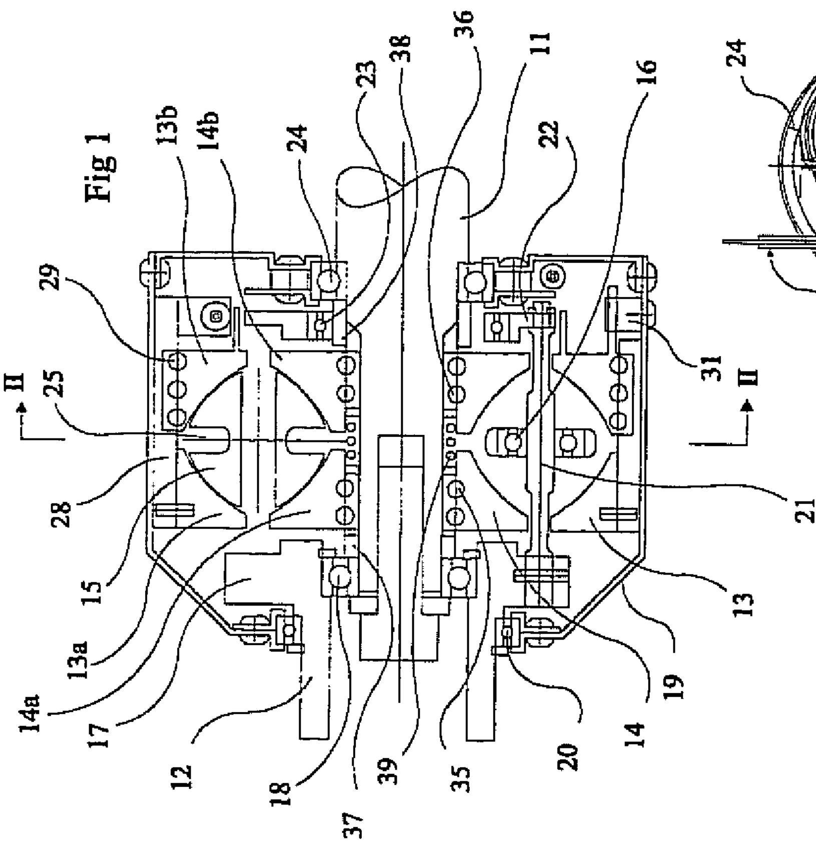


Fig 1

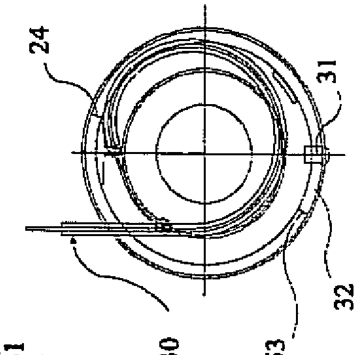


Fig 5

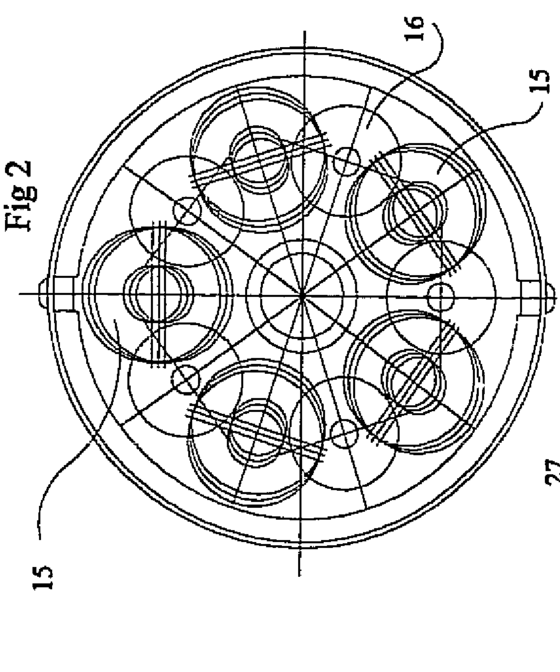


Fig 2

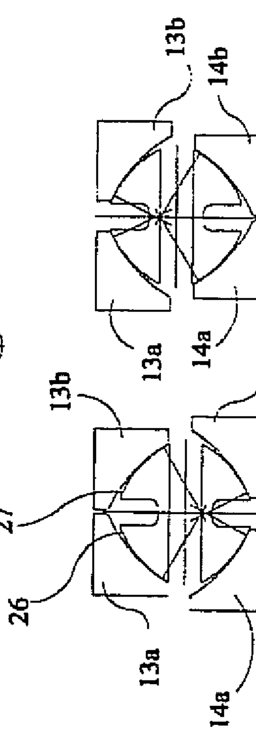


Fig 4

Fig 3

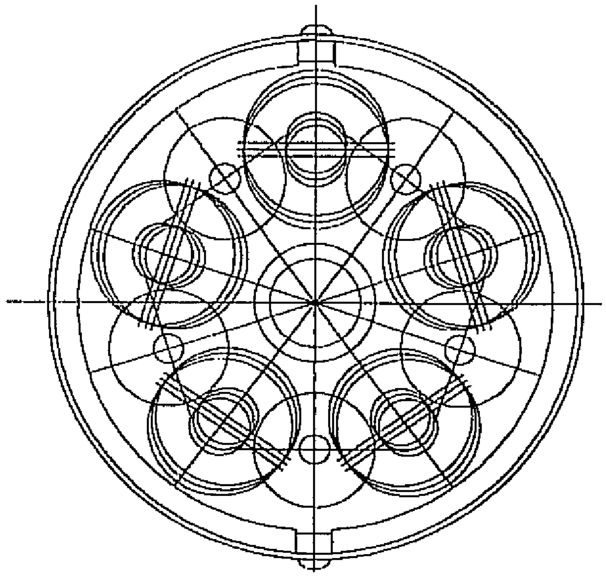


Fig 6

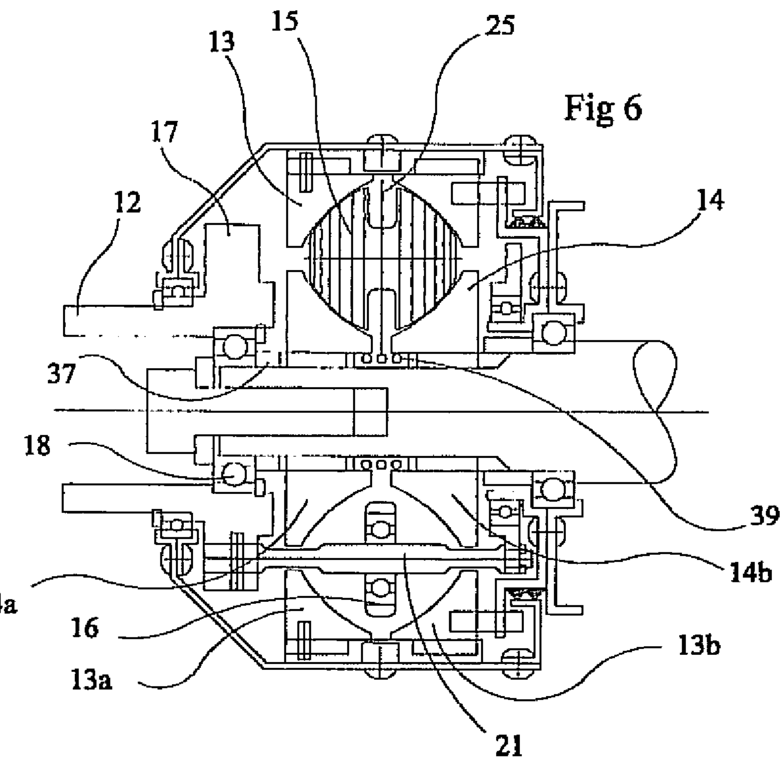
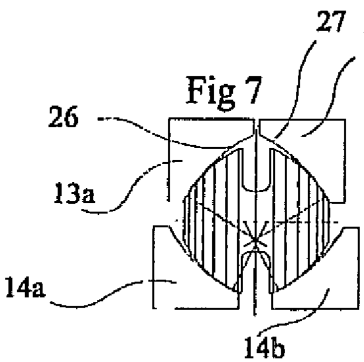
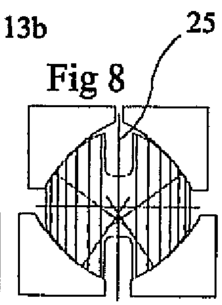


Fig 7



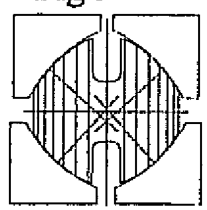
1st gear 0,14:1

Fig 8



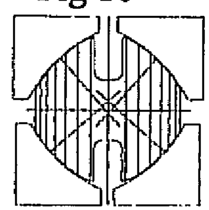
2nd gear 0,21:1

Fig 9



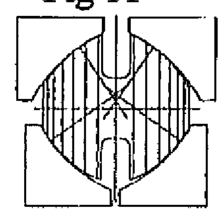
3rd gear 0,30:1

Fig 10



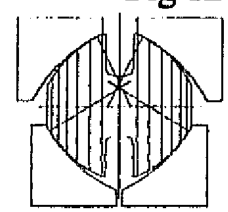
4th gear 0,40:1

Fig 11



5th gear 0,51:1

Fig 12



6th gear 0,62:1

A BI-DIRECTIONAL CONTINUOUSLY VARIABLE TRANSMISSION DEVICE

The present invention relates generally to a continuously variable transmission device, and particularly to such a device in which the forces are transmitted by rolling traction.

The present invention is an improvement in and development of the continuously variable transmission device described in the Applicant's earlier application number 9815952.8. That earlier application describes a continuously variable transmission device of the type having planet members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planet members in rolling contact therewith.

In that device there are provided means sensitive to the torque applied to a drive-transmitting member of the transmission, which is operable both to determine the compensating variation in the separation of the two parts of the other race and thus the transmission ratio of the device, and also to vary the forces exchanged between the planets and the races normal to the interface between them.

In that earlier application the drive transmission from an input shaft to an output shaft could only take place in one direction of rotation since the torque-sensing mechanism described, which in the specific embodiment described involved a helical interengagement between one of the two race parts in the "other" race and a cooperating component allowed the two race parts to be urged towards one another by the forces exerted on them in operation only when the direction of rotation of the input shaft corresponded to that of the helical interengagement. Relative rotation between the input shaft and the output shaft in the opposite direction would result in a relative separation of the "other" race parts which would effectively result in a reduction in the contact forces and, ultimately, to a decoupling of the input and output members. This, of course, has certain advantages in some circumstances, particularly where an over-run free-wheel effect is desirable. However, for use as a motor vehicle transmission, especially one in which engine over-run is used for braking, the free-wheel effect is unwanted and, indeed, decidedly undesirable.

The present invention seeks to provide a continuously variable transmission device of the type described in the Applicant's earlier patent application number 9815952.8 (the contents of which are incorporated herein by reference) in which the transmission of torque from an

input to an output shaft can take place in either direction of rotation.

According to one aspect of the present invention, therefore, there is provided a continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which there are provided means sensitive to the torque applied to a drive-transmitting member of the transmission device, operable both to determine the compensating variation in the separation of the two parts of the other race and thus the transmission ratio of the device and to vary the forces exchanged between the planets and the races normal to the interface between them, and in which the said torque-sensitive means include the two axially spaced, relatively moveable parts of the said other race, each said part being itself axially movable in two directional senses from a central position and engagable by limit stop means whereby to allow the transmission of rotary drive from a rotary drive input member to a rotary drive output member of the transmission device in each of two opposite senses of rotation.

In a preferred embodiment of the invention the said

relatively movable race parts of the torque-sensitive means are interconnected with the input drive member by a screw-thread engagement of the same hand by which rotary drive is transmitted when axial displacement of a race part is restrained.

The thread flights of the screw thread engagement are preferably interengaged by rolling elements such as balls although this is not essential. The provision of interengaging balls helps significantly to reduce frictional resistance in the device.

The said two relatively movable race parts of the torque-sensitive means may be oppositely axially resiliently biased. This resilient bias act to "prime" the torque-sensing reaction of the device and in a preferred embodiment of the invention the resilient biasing of the said two relatively movable race parts is achieved by a compression spring located between them.

Of course, in order to ensure that bi-directional rotation can take place each of the two race parts must ultimately be restrained from axial movement such that the other race part can, effectively "screw up" against it by the helical action exerted on it by the input member. Such limit stop means may comprise respective abutments on or carried by or associated with the said input drive member.

In one embodiment of the invention the two race parts of the said one race of the transmission device, the axial separation of which is selectively variable, are each carried on a casing of the transmission device in such a way as to have a limited rotational displacement in each of two opposite rotational senses. The relative axial separation of the two race parts of the said one race may be achieved by a helical interengagement of at least one of the two race parts with a fixed member of the transmission device, the two race parts both being relatively turnable with respect to the said fixed member. Such relative turning movement of the two race parts of the said one race may be achieved by any means which act directly between them rather than between one member and a fixed part. One means by which this can be achieved comprises a Bowden cable acting between the two race parts.

The present invention also comprehends, independently of the structure allowing bi-directional rotation to be achieved, a continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which the planetary members each have a circumferential annular groove the axis of

which substantially coincides with the respective rolling axis about which each planetary member turns as it rolls in contact with the races, the said annular grooves being engaged by roller follower members acting to guide the planetary members to maintain their orientation in their planetary motion.

This latter feature enables a greater load-carrying capacity to be achieved because a greater number of planetary members can be arranged in a given annular space because the circumferential space occupied by a planetary member can overlap that occupied by a planet follower.

The planet followers are preferably carried by a common carrier member through which drive transmission is conveyed to an output drive member of the device.

According to a further aspect of the present invention a continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, has planetary members each with arcuately curved surface portions in rolling contact with correspondingly curved portions of the respective races,

the radius of curvature of the said surface portions of the planetary members being greater than the effective radius of the planetary member itself.

This can be visualised by imagining the planetary members as spheres of a given diameter notionally split to remove a central portion and reassembled with the remaining quadrants in contact with one another. The radius of curvature of the surface portions will thus match that of the "original" sphere whilst the diameter of the newly-assembled sphere will be less than the diameter of the original sphere. Such planets may also be formed with circumferential grooves for receiving roller follower guide members as discussed above. There may further be provided means for guiding the planetary members to maintain the orientation of their rolling axes as they roll over the contacting surfaces of the races. Such guide members may be the above-mentioned rollers engaged in the circumferential grooves.

The purpose of enlarging the radius of curvature of the surface portions in relation to the diameter of the planetary member itself, is to extend the range of ratios which can be transmitted by the transmission device. In a specific embodiment, which will be described in more detail hereinbelow, the ratio range can be extended to 4.3:1.

In a further aspect of the present invention, which may

be considered independently of the other aspects described hereinabove, there is provided a continuously variable transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which each planetary member has a plurality of elementary annular contact surface portions having a substantially constant inclination to the rolling axis of the planetary member itself.

This allows the continuously variable transmission to be provided with preferred adjustment positions effectively representing specific gear ratios of a conventional gear box. Increased load-bearing capacity is also achieved by providing what amounts to a line rather than a point contact between the planets and the races over the surface portions having substantially constant inclinations.

This can be viewed as a planetary member having a generatrix which includes a section comprising a plurality of substantially rectilinear elementary portions. The races may have substantially continuously curved contact surfaces or may have respective contact surfaces for rolling contact with the planetary members,

each having correspondingly inclined elementary annular contact surface portions substantially matching those of the planetary members.

Various embodiments of the present invention will be more particularly described, by way of example, with reference to the accompanying drawings in which:

Figure 1 is an axial sectional view through a first embodiment of the present invention;

Figure 2 is a cross-sectional view taken on the line II-II of Figure 1;

Figures 3 and 4 are schematic detail views showing components of the embodiment of Figures 1 and 2 in two different operating configurations;

Figure 5 is a schematic cross-sectional view of the embodiment of Figure 1 showing the relative positions of an adjustment mechanism;

Figure 6 is an axial sectional view of an alternative embodiment of the invention; and

Figures 7 - 12 are schematic views of a detail of the embodiment of Figure 6 showing the components in different configurations for achieving different gear ratios.

Referring first to Figures 1 - 5 the device shown comprises a bi-direction continuously variable transmission device for transmitting rotary drive from an input shaft 11 to an output drive member 12 illustrated

as a tubular component to which, of course, an output drive shaft may be coupled by any known means.

The drive transmission device comprises inner and outer races 13, 14 each comprising axially spaced parts 13a, 13b; 14a, 14b between which roll planet members 15 circumferentially intercalated with roller follower members 16 carried on a common carrier 17 from which the output shaft 12 projects and which is borne on the input shaft 11 by a rolling element bearing 18 and on an outer casing 19 by a rolling element bearing 20.

The common carrier 17 has respective spindles 21 extending through and supporting the roller followers 16. Each spindle 21 is carried at its other end by a carrier plate 22 born on the input shaft 11 by a rolling element bearing 23. At this end the drive shaft 11 is born on the casing 19 by a rolling element bearing 24.

As can be seen in Figures 3 and 4, the planetary members 15 are generally spherical bodies divided into two axially separated parts by a circumferential annular groove or channel 25 into which the adjacent roller followers 16 engage in order to guide the planetary bodies 15 to turn about a rolling axis parallel to the axis of the drive shaft 11. Other than their engagement with the roller followers 16 and the races 13, 14 the planetary members 15 are unrestrained.

Contact between the planetary members 15 and the races 13, 14 takes place at two curved surface portions 26, 27 of the planetary body which, as will be appreciated from Figures 3 and 4 have a radius of curvature which is greater than the overall radius of the generally spherical body 15.

The radial position of the planetary body 15 is determined by the axial separation of the radially outer race parts 13a, 13b which axial separation is controlled by a screw threaded interengagement between the two race parts themselves, for which purpose the race part 13a is secured to a cylindrical sleeve 28 for rotation therewith. The screw threaded inter engagement of the two race parts is represented in Figure 1 of the drawings by the balls 29. A Bowden cable 30 (see Figure 5) is connected with its outer sheath engaging one of the two race members 13a, 13b and its inner cable engaged with the other such that axial forces applied between the sheath and the inner cable can cause relative turning motion of the race parts 13a, 13b. Depending on the direction of rotation of the shaft 11, this will result in axial displacement of the two race parts the rotation of which is limited by a stop 31 which engages in a recess 32 defined between end shoulders 33, 34 and a projecting head.

As will be appreciated from a consideration of Figures 3

and 4, relative approach of the two race parts 13a, 13b, as shown in Figure 3, will cause the planet member 15 to be urged radially inwardly towards the axis of the shaft 11, and this causes a corresponding separation of the parts 14a, 14b of the inner race 14. The forces exerted on the planet member 15 by the inner race 14 is generated by a torque-sensitive coupling comprising a screw threaded portion of the shaft 11 engaged in correspondingly threaded portions of the parts 14a, 14b, each of the same hand and represented in the drawing by the interconnection balls 35, 36.

Axial displacement of the inner race parts 14a, 14b is limited by abutment stops 37, 38 and a priming spring 39 urges the two race parts 14a, 14b apart. Thus, depending on the direction of rotation of the shaft 11, one or other of the race parts 14a, 14b will be limited in its axial displacement by the respective axial abutment shoulder 37, 38 such that the screw-turning motion imparted to the other by the rotation of the shaft 11 will compensate the forces exerted by the choice of axial separation of the two outer race parts 13a, 13b. As illustrated in Figure 3, with the two parts 13a, 13b closely together, the planet member 15 is urged radially inwardly such that the inner race parts 14a, 14b are urged apart so that the rolling contact of the planet member between the inner and outer races results in a low ratio in the region of 0.14:1. When the outer race parts

13a, 13b are allowed to separate by action of the cable 30 reducing the tension between the inner cable and outer sheath, the torque exerted by the shaft 11 will cause the inner race parts 14a, 14b to move towards one another increasing the transmission ratio up to a maximum of 0.62:1 as illustrated in Figure 4. This ratio range is increased by the enlargement of the radius of curvature of the contacting surfaces, 26, 27 of the planet 15 in relation to the overall general diameter of the planet itself. The load-bearing capacity of the transmission is also increased by the presence of the channels 25 in the planets which allows a greater number of planets to be arranged within a transmission casing of given size. As illustrated in Figure 2 it will be seen that there are five planet members in the array, intercalated with five roller followers each carried on a respective spindle 21. The effective diameter is determined by the need for the presence of the spindles 21 to transmit forces from the roller followers to the carrier. Moreover, by mounting the inner race parts 14a, 14b on a common thread axially compressive forces can be generated regardless of the direction of rotation of the drive shaft 11 as, in each case, the "trailing" race part will be urged towards the other when this contacts its respective abutment.

Turning now to Figure 6 there is shown a transmission device in which, although still notionally continuously variable, will act to provide a number of preferential

gear ratios at which the device will stop in the absence of overriding forces. The general configuration of the device illustrated in Figure 6 is similar to that of Figure 1 and, therefore, the same or corresponding components will not be described again. In this embodiment the planet members 15 have contact surfaces 26, 27 composed of a plurality of annular conical surfaces having a linear generatrix to form effectively annular "facets" which therefore effectively define a given gear ratio when in contact with the corresponding contact surfaces of the race parts. Figures 7 to 12 illustrate the relative positions of the inner and outer race parts for the six gear ratios determined by the six annular facets of the planet members in this embodiment.

CLAIMS

1. A continuously variable drive transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which there are provided means sensitive to the torque applied to a drive-transmitting member of the transmission operable both to determine the compensating variation in the separation of the two parts of the other race and thus the transmission ratio of the device and to vary the forces exchanged between the planets and the races normal to the interface between them, in which the said torque-sensitive means include the two axially spaced, relatively movable parts of the said "other" race, each said part being itself axially movable in two directional senses from a central position and engagable by limit stop means whereby to allow the transmission of rotary drive from rotary drive input member to a rotary output member of the transmission device in each of two opposite senses of rotation.

2. A continuously variable drive transmission device as claimed in Claim 1, in which the said relatively movable race parts of the torque-sensitive means are

interconnected with the input drive member by a screw-thread engagement of the same hand by which rotary drive is transmitted when axial displacement of a race part if restrained.

3. A continuously variable drive transmission device as claimed in Claim 2, in which the thread flights of the screw-thread engagement are interengaged by rolling elements such as balls.

4. A continuously variable drive transmission device as claimed in any of Claims 1 to 3, in which the said two relatively variable race parts of the torque-sensitive means are oppositely axially resiliently biased.

5. A continuously variable drive transmission device as claimed in Claim 4, in which the resilient biasing of the said two relatively movable race parts is achieved by a compression spring located between them.

6. A continuously variable drive transmission device as claimed in any preceding claim, in which the said limit stop means comprise respective abutments on or carried by or associated with the said input drive member.

7. A continuously variable drive transmission device as claimed in any preceding claim, in which the two race parts of the said one race of the transmission device the

axial separation of which is selectively variable, each carried on a casing of the transmission device in such a way as to have a limited rotational displacement in each of two opposite rotational senses.

8. A continuously variable drive transmission device as claimed in Claim 7, in which the relative axial separation of the two race parts of the said one race are achieved by a helical interengagement of at least one part of the two race parts with a fixed member of the transmission device, the two race parts both being relatively turnable with respect to the said fixed member.

9. A continuously variable drive transmission device as claimed in Claim 8, in which the relative turning of the two race parts of the said one race is achieved by means of a Bowden cable acting between them.

10. A continuously variable drive transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which the planetary members each have a circumferential annular groove the axia of which substantially coincides with the respective rolling

axis about which each planetary member turns as it rolls in contact with the races, the said annular grooves being engaged by a roller follower members acting to guide the planetary members to maintain their orientation in their planetary motion.

11. A continuously variable drive transmission device as claimed in Claim 10, in which the planet follower members are carried by a common carrier member through which drive transmission is conveyed to an output drive member of the device.

12. A continuously variable drive transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which the planetary members each arcuately curved surface portions in rolling contact with the corresponding curved portions of the respective races, and the radium of curvature of the said surface portions of the planetary members is greater than the effective radius of the planetary member itself.

13. A continuously variable drive transmission device s claimed in Claim 12, in which there are further provided means for guiding the planetary members to maintain the

orientation of their rolling axis as they roll over the contacting surfaces of the races.

14. A continuously variable drive transmission device of the type having planetary members in rolling contact with radially inner and outer races each comprising two axially spaced parts, with control means for selectively varying the axial separation of the two parts of one race and thus the radial position of the planetary members in rolling contact therewith, in which each planetary member has a plurality of elementary annular contact surface portions having a substantially constant inclinations to the rolling axis of the planetary member itself.

15. A continuously variable drive transmission device as claimed in Claim 14, in which the generatrix of each planetary member includes a section comprising a plurality of substantially rectilinear elementary portions.

16. A continuously variable drive transmission device as claimed in Claim 14 or Claim 15, in which the races have respective substantially continuously curved contact surfaces.

17. A continuously variable drive transmission device as claimed in Claim 14 or Claim 15, in which the races have respective contact surfaces for rolling contact with the

planetary members, having correspondingly inclined elementary annular contact surface portions substantially matching those of the planetary members.

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Application No:	GB 9818557.2	**Examiner:**	Mike Mckinney
Claims searched:	1 to 9	**Date of search:**	25 January 2000

Patents Act 1977
Search Report under Section 17

Databases searched:

UK Patent Office collections, including GB, EP, WO & US patent specifications, in:

UK Cl (Ed.R): F2D (DEJ)

Int Cl (Ed.7): F16H 15/50

Other: ONLINE: WPI; EPODOC; JAPIO.

Documents considered to be relevant:

Category	Identity of document and relevant passage		Relevant to claims
X	GB 0821860	(ROLLER GEAR) see fig 1, lines 100 to 120 page 2 and lines 61 to 71 page 3.	1 to 8.
X	GB 0769583	(OTTO SINGER) see fig 1 and lines 36 to 45 page 3.	1 to 8.
X	GB 0702761	(ROLLER GEAR) see fig 1 and lines 98 to 113 page 2.	1 to 8.

X	Document indicating lack of novelty or inventive step	A	Document indicating technological background and/or state of the art.
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Application No:	GB 9818557.2	**Examiner:**	Mike Mckinney
Claims searched:	10 and 11.	**Date of search:**	25 January 2000

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Databases searched:

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Other: ONLINE: WPI; EPODOC; JAPIO.

Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
	NONE	

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Application No:	GB 9818557.2	**Examiner:**	Mike Mckinney
Claims searched:	12 to 17.	**Date of search:**	25 January 2000

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Documents considered to be relevant:

Category	Identity of document and relevant passage	Relevant to claims
X	US 1585140 (ERBAN) see fig 6 and lines 99 to 113 page 2.	12 to 17.

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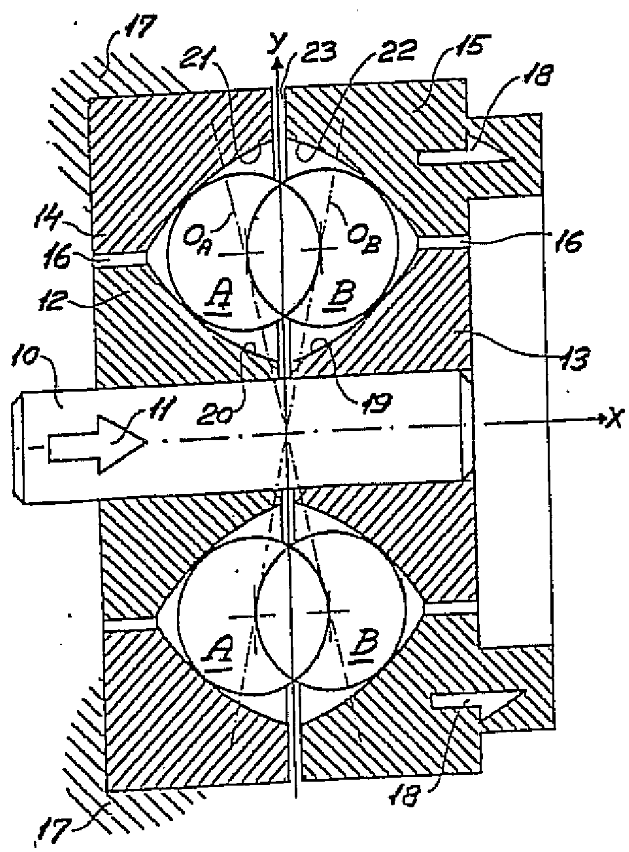
(57) Abstract

A steplessly variable ball gear has two co-axial rings of balls (A and B), which are so supported by two inner and two outer rolling paths (19-22) formed on moulding rings (12-15) that each ball in one ring touches two rolling paths and two balls in the other ring. The inner and outer rings are axially shiftable with respect to each other. As the distance between both the inner and the outer rings is constant and the rolling paths have concave-curved, preferably circular arc-shaped cross-sections, it can be ensured that the contacts of tangency, required for transmission of torque, between the balls and between the balls and the rolling paths are maintained over a wide variation range.



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A transmission mechanism

The invention relates to a transmission mechanism of the type defined in the introductory portion of claim 1. The axial shiftings of one or the other set of rolling paths must result in such changes of the mutual positions of the balls and the rolling paths that the transmission ratio of the mechanism continuously changes with the shiftings.

The US Patent Specification 2 862 407 discloses a transmission mechanism of the present type in various embodiments, which share the feature that one set of rolling paths has rectilinear generatrices, while a single one has convex generatrices. For the balls in such a construction to maintain the four contacts of tangency with a changed transmission ratio, it is necessary that either the two inner or the two outer rolling paths are axially shiftable with respect to each other, and that the shiftable rolling path is under the action of a spring force. As the tangential forces forming part of the torques transmitted by friction roll gears are the normal force multiplied by the coefficient of friction, the normal forces must be of a size ten times the force which multiplied by the power arm constitutes the moment transferred.

Moreover, with the generatrix geometry of the rolling paths used in this known friction roll gear only a very limited variation range can be obtained, and consequently most practical uses will require multiplication of the transmission ratio, e.g. by combining the gear with a planet gear, as is also shown in the patent specification.

The invention consists in the finding that the establish-

ment of a constant axial distance between both the inner and the outer rolling paths makes it possible, on the basis of simple considerations and desired data, to calculate such dimensions and shapes of the constituent elements as will allow the contacts of tangency required for transmission of torque to be maintained within a variation range which makes the transmission mechanism generally applicable for the performance of transmission tasks in practice.

When the mechanism is constructed as stated in claim 2 both the rotation of the balls about their individual axis of rotation and their rotation about the main axis are utilized for transmission. The circumstance that the individual axes of rotation always intersect the rolling paths or imaginary inwardly extending extensions of them within the ball points of tangency ensures that the balls will not assume a position in which they only rotate about fixed individual axes of rotation.

There are many curve shapes which can satisfy the necessary conditions, but it is preferred in practice to use rolling paths having substantially circular arc-shaped generatrices, as stated in claim 3, because such rolling paths are relatively easy to manufacture and can meet all normal requirements of transmission ratios and variation widths.

When all the balls are of the same size, the rolling paths in the transmission mechanism of the invention have generatrices which are symmetrical in pairs, as stated in claim 4. The greatest variation width can be obtained when the four rolling paths have double-symmetrical generatrices, as stated in claim 5.

The invention will be explained in greater detail below

with reference to the drawing, in which

figs. 1 and 2 show an axial section of an embodiment of the ball gear of the invention in a neutral position and an extreme position, respectively,

fig. 3 shows the balls in such a gear, as seen in the direction of the gear axis,

figs. 4 and 5 are geometrical figures serving to explain the operation of the gear,

fig. 6 is a horizontal section through the centres of the two lower balls in fig. 3,

figs. 7-13 show geometrical figures serving to explain the operation of the gear,

fig. 14 is a schematic axial section of the upper half of a gear like the one shown in fig. 1,

fig. 15 show symbols used in fig. 16,

figs. 16a and b are a survey of various gear types of the invention,

fig. 17 is an axial section like the one shown in fig. 2, but with the gear in the opposite extreme position,

figs. 18 and 19 are spatial views serving to illustrate the radii used in the calculation of the transmission ratio of the gear,

fig. 20 schematically shows a development of a part of the ball gear and serves to illustrate directions of rotation,

figs. 21-27 are geometrical figures used in the calculation of the transmission ratio,

figs. 28 and 29 show the two ball rings in a gear of the invention in a limit position as seen from the side and in the direction of the gear axis, respectively,

figs. 30-32 are geometrical figures serving to illustrate certain limits of gears of the invention, and

figs. 33-38 are schematic axial sections of various variants, or parts of them, of the ball gear of the invention.

The transmission mechanism shown in figs. 1 and 2 has an input shaft 10 which receives power marked by an arrow 11 and mounts two moulding rings 12 and 13. Two other moulding rings 14 and 15 are mounted around and coaxially with the rings 12 and 13 and are spaced from these in a radial direction by ring-shaped spaces 16; one 14 is stationary as shown by hatched areas 17, while the other 15, spaced from the first one in an axial direction by a space 23, is freely rotatable but not slidable axially and forms the output element from which the power marked by two half-arrows 18 is taken. The four moulding rings 12, 13, 14 and 15 are formed with rolling paths 19, 20, 21 and 22, respectively, which face toward each other and have circular arc-shaped cross-sections in the embodiment shown. Two ball rings are so mounted between the rolling paths that each ball A in one ring touches the inner rolling path 20, the outer rolling path 21 and two balls B in the other ring, and similarly each of these balls B touches the inner rolling path 19, the outer rolling path 22 and two balls A in the first ring.

The frictional contacts, as caused by the rotation of

the input shaft 10, between the rolling paths and the balls and between the balls will make the balls rotate about their own individual axes O_A and O_B and about the gear axis x of the entire mechanism. The rotation of the ball ring about this axis takes place in the opposite direction of that of the input shaft. The axes of rotation O_A and O_B of the balls intersect each other on the gear axis x.

In fig. 1 there is plotted an axis y which is at right angles to the axis x and is an axis of symmetry for the two axially fixed outer rolling paths 21 and 22.

Axial shifting of the inner moulding rings 12 and 13, as shown in fig. 2, changes the quantities which determine the ratio of the rotary speed of the input shaft 10 to that of the output element 15, i.e. the gear ratio of the transmission mechanism. During such shifting the contacts of tangency between the rolling paths and the balls and between the balls must be maintained, and it will be explained below what conditions are to be met in order for this to be possible.

Fig. 3 shows two ball rings each having three balls A and B, as seen in the direction of the gear axis. In addition to the y axis an axis z is plotted in fig. 3 which is at right angles to both axes x and y. Moreover, equiangular lines P are plotted, representing planes which intersect each other in the x axis and in which the ball centres move with relative axial movements of the ball rings. One of the lines P coincides with the z axis. The angle between two adjacent planes P, hereinafter called the centre angle, is designated β and depends upon the number n of balls:

$$\beta = \frac{360}{n} \qquad (1)$$

If the balls in one ring are shifted towards the x axis, the balls in the other ring will move away from the x axis, as shown by broken lines. The projection of a line or an imaginary rod S between the centres of two balls touching each other on a line perpendicular to the bisector plane of the centre angle remains constant, as readily appears from the two congruent triangles resulting from the projection of the broken line S on the solid line S. As the balls must constantly touch each other, the direct distance, as represented by the length of the rod S, between the ball centres must constantly be equal to 2 R_k, where R_k is the radius of the balls.

If the two planes P in which the ball centres can move had been parallel with a distance of less than 2 R_k, the possibilities of motion for the ball centres while maintaining contact of tangency between the balls might be represented by two circles Cy of the same size, fig. 4, which are so disposed in their respective ones of the two planes as to form end faces of an imaginary cylinder Cy, whose axis is perpendicular to the planes and in which the imaginary rod S will extend between an arbitrary point on one circle and the diametrically opposite point on the other circle. The point of tangency of the balls will be the centroid of this cylinder.

In reality, however, the two planes in which two adjacent ball centres move are not parallel, but form the angle β with each other. In fig. 5 the imaginary cylinder Cy and rod S are plotted in three different positions between two planes P. The position shown by solid lines is the neutral position in which the two ball centres are equidistantly spaced from the x axis, and the position shown by broken lines is an extreme position in which the difference between the distances of the ball centres

from the x axis are as great as possible. The position shown by broken lines is an intermediate position. Above the cylinder are shown the projections of the rod on the z axis in three positions. It will be seen that relative movements of the ball centres are reflected by axial movements of the cylinder Cy, and that the paths in which the ball centres can move in the planes P are the ellipses in which the cylinder face intersects the planes. One of these ellipses is shown at E in fig. 5 together with a circle Ci representing the cylinder, as seen from the end. Half the minor axis of the ellipse which is equal to the radius in the circle, is designated a and half its major axis is designated b.

In fig. 3 two adjacent balls A and B are hatched. Suppose a section is made through the centres of these balls in parallel with the xz plane, and the result will be a view like the one shown in fig. 6 where the two ball radii R_k disposed in elongation of each other represent the imaginary rod, which forms an angle α with a line which is parallel with the z axis and extends through the centre of the ball A and which represents an outer generatrix of the imaginary cylinder, as seen in the direction of the x axis. The end face of the cylinder is represented by a line of the length 2a which extends through the centre of the ball B and is parallel with the x axis. The angle α depends upon the "density" of the balls in the neutral position. It will be seen from figs. 5 and 6 that the axes of the ellipse can be expressed by

$$a = R_k \cdot \sin\alpha \qquad (2)$$

$$b = \frac{a}{\cos(\beta/2)} = \frac{R_k \cdot \sin\alpha}{\cos(\beta/2)} \qquad (3)$$

In practice, the shifting of the balls is produced by axial shifting of one set of moulding rings with respect to the other set. This implies that the imaginary cylinder Cy is shifted in a direction perpendicular to its axis and in parallel with the gear axis x. Thus the ball centres perform a composite movement with a radial and an axial component along a resulting path which defines the cross-section of the rolling paths in planes containing the gear axis. Theoretically, an infinite number of curve shapes may be used; but only one curve shape satisfies the demand that for the achievement of the greatest possible variation of the force arms determined by the location of the points of tangency of the balls with the rolling paths, there must be symmetrical movement both radially and axially of these points of tangency. In view of the double-symmetrical movement of the points of tangency of the balls with the rolling paths, the relation between the oppositely directed radial shifting and common axial shifting of two ball centres must necessarily be so that the shifting paths of the ball centres are likewise symmetrical.

The required conditions of symmetry are met by the cycloid. A cycloid produced by rolling a circle having a radius a on a straight line is shown in fig. 7. The oppositely directed movements of the ball centres along the edges of the imaginary cylinder and their common axial shifting together with the cylinder can be depicted by two half cycloids in the bisector plane of the centre angle, as shown in fig. 8, where the axis parallel with the x axis and perpendicular to the cylinder axis is designated x'. In the range $a \geqq y \geqq -a$, where y is taken from the x' axis, the curves can be expressed by the equation

$$x = \sqrt{a^2-y^2} + \frac{\left[\sin^{-1}\left(\frac{y}{a}\right)\right] \cdot \pi \cdot a}{180}$$

The ball centres can be fixed on these uniform curves symmetrical about the y axis by means of identical curves, which are symmetrical with the first ones with respect to the x' axis, by shifting one pair of curves in relation to the other pair along the x' axis, as shown by broken lines in fig. 9. These double-symmetrical curves establish the geometrical basis for the rolling path profiles. However, the curves are plotted on a plane forming an angle of half the centre angle β with the planes P in which the ball centres can move. The curves must therefore be projected on one of these planes for them to form a basis for the desired profiles. The y co-ordinates of the curves must therefore be divided by the cosine to half the centre angle, resulting in the view shown in fig. 10. In the range $c \geq y \geq -c$, where c is the selected value of the maximum shifting in a radial direction, i.e. in the direction of the y axis, the curves can now be expressed by

$$x = \sqrt{a^2-(y \cdot \cos \beta/2)^2} + \frac{\left[\sin^{-1}\left(\frac{y \cdot \cos \beta/2}{a}\right)\right] \cdot \pi \cdot a}{180}$$

As will appear from the following, the gearing range of the transmission mechanism is determined by the maximum difference obtainable between the radii from the gear axis to the points of tangency between the balls and the rolling paths and the radii from the individual axes of rotation of the balls to the same points of tangency as well as between the radii from the latter axes to the mutual point of tangency of the balls, by shifting of the balls of the ball rings. As the maximum differences in radii are determined not only by the number of balls,

the angle α and the value of c, but also by the travel radius r of the fundamental cycloid, the radius r may be used in the adaption of the gearing range, as desired. The travel radius of the cycloid is defined by

$$r = \frac{\left(\sqrt{a^2-(c\cdot\cos\beta/2)^2} - m\right) \cdot 180}{\left[\sin^{-1}\left(\frac{c\cdot\cos\beta/2}{a}\right)\right]\cdot\pi}$$

where m is the y value corresponding to x' = -c, so the generalized formula of the shifting curve of the ball centres in the range $c \geqq y \geqq -c$ will be:

$$x = \sqrt{a^2-(y\cdot\cos\beta/2)^2} + \frac{\left[\sin^{-1}\left(\frac{y\cdot\cos\beta/2}{a}\right)\right]\cdot\left(\sqrt{a^2-(c\cdot\cos\beta/2)^2} - m\right)}{\sin^{-1}\left(\frac{c\cdot\cos\beta/2}{a}\right)}$$

Fig. 11 shows an example of a travel radius r smaller than the radius a of the imaginary cylinder.

The curve shapes found here lead to rolling path profiles which are rather difficult to manufacture; but the curve segments which are needed as a basis for the profiles are very approximate to circular arcs which lead to profiles that lend themselves to manufacture. It can be calculated that the maximum deviation is an order less than the inaccuracy which is usually incident to the balls. To this should be added that the effects of the curve inaccuracies on two touching balls virtually neutralize each other because of the curve symmetry.

Fig. 12 shows a view corresponding to fig. 10, where the ball centre paths are circular arcs Ci_1 and Ci_2 with radius R and with their respective centres C_1 and C_2 at the same distance y_c from the minor axis or the x' axis

and at the same distance x_c from and at their respective sides of the y axis, which coincides with the major axis in the solid ellipse E representing the imaginary cylinder in its neutral position. Moreover, the simplification has been made in fig. 11 that the broken ellipse representing an extreme position of the imaginary cylinder intersects the y axis at the points $y = \pm c$. In the neutral position the projection on the x' axis of the imaginary rod S forms the minor axis of the ellipse. The projection of the rod on the x' axis in the extreme position of the cylinder is designated d.

If the shifting takes place to the left instead of to the right in fig. 12, the ball centres will follow extensions of the circular arcs, viz. in respect of the arc Ci_1 down to the line $y = -c$ which it intersects on the y axis, and in respect of the arc Ci_2 up to the line $y = c$.

If the right ball centre, fig. 12, during shifting of the ellipse to the right, moves downwardly instead of upwardly as shown and in case of shifting to the left moves upwardly, two other arc-shaped paths Ci_3 and Ci_4 result, forming a mirror picture of the arcs Ci_1 and Ci_2, as shown in fig. 13. During relative movements one set of arcs is shifted in the direction of the x' axis with respect to the other. The ball centres will constantly lie in their respective intersections between the upper and the lower arc paths. It will be seen from fig. 13 that the maximum relative shifting of the set of arcs is equal to d. On the basis of the ellipse formula d can be expressed by the ellipse axes and the quantity c:

$$d = 2a\sqrt{1 - c^2/b^2}$$

The following three equations with the three unknown

quantities R, x_c and y_c may be deduced from fig. 12.

$$R^2 = (y_c + c)^2 + x_c^2$$

$$R^2 = y_c^2 + (x_c + a)^2$$

$$R^2 = (y_c - c)^2 + (x_c + d)^2$$

Hence the coordinates of the ball centre C_1 and the radius of the circular arc Ci_1

$$x_c = \frac{2a^2 - 2c^2 - d^2}{(ad - 4a)} \qquad (4)$$

$$y_c = \frac{2x_c \cdot d + d^2}{4c} \qquad (5)$$

$$R = \sqrt{(y_c + c)^2 + x_c^2} \qquad (6)$$

When the rings are formed with a track radius

$$R_i = R + R_k \qquad (7)$$

as shown in fig. 14, shifting of the inner and outer moulding rings with respect to each other will produce the desired elliptical path movement of the ball centres and thus ensure that the contact of tangency between the balls and the rolling paths and between the balls is maintained.

In fig. 14 is marked the radius R_y of the gear bearing, as is the case with fig. 5, too. Half the length of the projection of the imaginary rod shown by a solid line in fig. 3 is $R_k \cdot \cos \alpha$, and hence

$$R_y = \frac{R_k \cdot \cos\alpha}{\sin(\beta/2)} \qquad (8)$$

All dimensions of the transmission mechanism can be calculated from the foregoing equations.

Ball gear types

In the embodiment of the transmission mechanism of the invention shown in figs. 1 and 2 the moulding ring 14 is, as mentioned, fixed, while the input power is fed to the moulding rings 12 and 13, and the output power is taken from the moulding ring 15. The use of four moulding rings, however, may be combined in many other ways to provide various transmissions and variation ranges. Some types may also be used as a differential. Figs. 16a and 16b are symbolic views of a plurality of different combinations with type designations, and fig. 15 shows the symbols used in these views. In fig. 15, 1, 2, 3 and 4 represent the four moulding rings, and in the type designations the figure before the hyphen represents the moulding ring or rings coupled to the input shaft, while the figure after the hyphen represents the moulding ring or rings coupled to the output shaft. The two types which have a slanted stroke in the type designation are differential structures, and the two figures separated by the slanted stroke represent the rings coupled to their respective output shafts.

Calculation of gear ratio

Fig. 17 shows a ball gear of type 14-3 in an extreme position corresponding to a shifting d of the moulding rings 1 and 4, or - with the designations used in figs. 1 and 2 - 12 and 13. In this position the radii A_c and B_c of the centre paths of the ball rings are

$$A_c = R_y - c \tag{9}$$

$$B_c = R_y + c \tag{10}$$

A comparison between figs. 14 and 17 moreover shows that the radii in the circular paths in which the ball rings touch the rolling paths are

$$A_i = A_c - R_k \cdot \frac{y_c + c}{R} \tag{11}$$

$$B_i = B_c - R_k \cdot \frac{y_c - c}{R} \tag{12}$$

$$A_y = 2\ R_y - B_i \tag{13}$$

$$B_y = 2\ R_y - A_i \tag{14}$$

A general idea of the meaning of the designations now introduced can be obtained by consideration of the spatial view shown in fig. 18. This figure additionally comprises the new designations a_i and a_y for the perpendicular distances from the points of tangency of the ball A with the rolling paths to the axis of rotation O_A of the ball, and b_i and b_y for the corresponding distances for the ball B. Also introduced are the designations a_k and b_k for the perpendicular distances from the common point of tangency of the balls to their respective axes of rotation.

Fig. 19 illustrates the same as fig. 18, but differently, because instead of the two balls it shows the planes P in which the ball centres can move. In fig. 19 are marked the regulating length d of the ball gear and the distance x from the centre of the double ball ring to the point where the axes of rotation of the balls intersect the x axis. The two axes of rotation must necessarily intersect each other on the x axis, as there must be no slipping

between the balls or between the balls and the rolling paths.

To calculate the gear ratio of a gear it is necessary to know all the quantities shown in fig. 19. The transmission possibilities of a gear can be determined on the basis of the conditions in the extreme positions between which it can be varied in stepless manner.

As mentioned before, the combinations shown in fig. 16 give various possibilities of transmission. For an arbitrarily selected combination it is possible to set up just as many equations as there are unknown radii on the basis of the geometrical conditions applying to the radii in the planes shown in fig. 19, and the gear ratio can thus be calculated. Here, there is selected the combination shown in fig. 17 with the two inner moulding rings 1 and 4, fig. 15, secured to the input shaft, one 2 of the outer moulding rings stationary and the other outer moulding ring 3 secured to the output shaft. This combination has the type designation 14-3. Fig. 1 shows the same gear in a neutral position, which is neutral gear, it being readily apparent for reasons of symmetry that the outer moulding ring must stand still just like the fixed moulding ring. It will be seen from fig. 20 that in operation the balls in the two rings always rotate oppositely each other irrespective of gear ratios. Both in figs. 17 and 19, the inner and the outer moulding rings are shifted the maximum distance d in one direction with respect to each other, corresponding to maximum transmission ratio.

On the basis of fig. 18 it is possible to set up a mutual dependence between some of the radii. Suppose the moulding ring 1 has rotated precisely one revolution, a point on the ball A will have moved the distance $2 \cdot \pi \cdot A_1$, which

in turn means that it has rotated

$$\frac{2\cdot\pi\cdot A_i}{2\cdot\pi\cdot a_i} = \frac{A_i}{a_i}$$ times about its own axis of rotation.

This rotation of the ball A has caused the point of tangency with the ball B to have moved the distance

$$\frac{A_i}{a_i} \cdot 2\,\pi\cdot a_k.$$

As the moulding ring 4 is secured to the moulding ring 1, this ring will likewise have moved one revolution, and the same considerations can now be made in respect of the ball B. On the ball B the point of tangency with the ball A must have moved the distance

$$\frac{B_i}{b_i} \cdot 2\cdot\pi\cdot b_k.$$

As the two distances must be of the same length, we have

$$\frac{A_i}{a_i}\, a_k = \frac{B_i}{b_i}\, b_k \qquad (15)$$

As mentioned, selection of another combination of fixed and rotating rings will involve another dependence between the radii than (15).

The task is now to find a distance x, fig. 19, which satisfies this equation. A_i and B_i are known from the equations (11) and (12) and what remains is thus to find a_i, a_k, b_i and b_k. These quantities can be found by geometrical considerations of figs. 18 and 19. a_i can be calculated by consideration of the plane shown in fig. 21 and containing the x axis and the axis of rotation A of the ball A. On the basis of this figure, and with l_A being

the distance along the axis O_a betwen its intersection with the x axis and the centre of the ball A, we have

$$l_A = \sqrt{A_c{}^2 + (x + d/2)^2} \tag{16}$$

$$v = 90 - \sin^{-1}\left(\frac{A_c - A_i}{R_k}\right) \pm \cos^{-1}\left(\frac{A_c}{l_A}\right) \tag{17}$$

$$a_i = R_k \sin(v) \tag{18}$$

It will be seen that all that is needed for a_i to be calculated is knowledge of x.

Similarly, a set of equations may be set up for the calculation of b_i on the basis of fig. 22, where the distance along the axis of rotation of the ball B from its intersection with the x axis to the centre of the ball is designated l_B:

$$l_B = \sqrt{B_c{}^2 + (x - d/2)^2} \tag{19}$$

$$v = 90 - \sin^{-1}\left(\frac{B_c - B_i}{R_k}\right) \pm \cos^{-1}\left(\frac{B_c}{l_B}\right) \tag{20}$$

$$b_i = R_k \cdot \sin(v) \tag{21}$$

b_i can now be calculated if x is known.

It will be seen from fig. 19 that the perpendicular distance S_1 from the x axis to the mutual point of tangency of the balls can be calculated from fig. 23:

$$S_1^2 = \frac{A_c^2 + B_c^2}{4} + \frac{A_c B_c}{2} \cdot \cos\beta \tag{22}$$

Then, the length T, as indicated in figs. 19 and 20, from the mutual point of tangency of the balls to the intersection of the axes O_A and O_B with the x axis can be calculated:

$$T = \sqrt{S_1^2 + x^2} \qquad (23)$$

Finally, the plane of fig. 25 is considered, containing the axes of rotation of the two balls. From this figure are obtained

$$a_k = T \cdot \sin\left(\cos^{-1}\left(\frac{1_A^2 + T^2 - R_k^2}{21_A T}\right)\right) \qquad (24)$$

$$b_k = T \cdot \sin\left(\cos^{-1}\left(\frac{1_B^2 + T^2 - R_k^2}{21_B \cdot T}\right)\right) \qquad (25)$$

A set of equations has now been formed which makes it possible to calculate x. These calculations are admittedly rather difficult, but can be made relatively easily by means of an electronic calculating machine.

When x has been found, the values of a_i, b_i, a_k and b_k can be found. What remains is then the calculation of A_y, a_y, B_y and b_y before all the radii of the gear are known. With x being now known, these calculations can be made from figs. 26 and 27:

$$v = 90 - \sin^{-1}\left(\frac{A_y - A_c}{R_k}\right) \pm \cos^{-1}\left(\frac{A_c}{1_A}\right) \qquad (26)$$

$$a_y = R_k \sin(v) \qquad (27)$$

$$v = 90 - \sin^{-1}\left(\frac{B_y - B_c}{R_k}\right) \pm \cos^{-1}\left(\frac{B_c}{1_B}\right) \qquad (28)$$

$$b_y = R_k \sin(v) \qquad (29)$$

As all radii in the extreme position of a selected

combination are now known, it is possible to calculate the maximum gear ratio of the gear.

Suppose that the ball ring A, figs. 17 and 18, has moved precisely one revolution about the gear axis, the point of tangency with the moulding ring 2 will have moved the distance $2\cdot\pi A_y$, and the ball will thus have moved

$$\frac{2\cdot\pi A_y}{2\pi a_y} = \frac{A_y}{a_y}$$ times about its own axis of rotation as moulding ring 2 is fixed.

A similar consideration shows that the moulding ring 1 must have moved

$$\frac{A_y a_i}{a_y A_i} - 1$$ times about its own axis of rotation, which is the gear axis; here one revolution which the ball ring has moved in the opposite direction must be subtracted.

The ball row B must similarly have moved one revolution. It appears from fig. 18 that a ball in this ring must have moved

$$\frac{A_y a_k}{a_y b_k}$$ times about its own axis of rotation, and hence moulding ring 3 must have moved

$$\frac{A_y a_k b_y}{a_y b_k B_y} - 1$$ times about its own axis of rotation which is the gear axis. Here too, one revolution which the ball

ring B has moved in the opposite direction must be subtracted.

It is known how many revolutions the moulding rings 1 and 4 (the input shaft) and the moulding ring 3 (the output shaft) move when the ball rings rotate just once about the gear axis. When the number of revolutions of the output shaft is 1, the number of revolutions of the input shaft can be calculated as

$$\frac{\frac{A_y a_i}{a_y A_i} - 1}{\frac{A_y a_k b_y}{a_y b_k B_y} - 1}$$

Rewritten, the gear ratio of the gear can be expressed as

$$1 : \frac{\frac{a_y}{A_y} - \frac{a_i}{A_i}}{\frac{a_y}{A_y} - \frac{a_k b_y}{b_k B_y}} \qquad (30)$$

As appears from the position in which the gear bearing is shown in figs. 17 and 18, and likewise from the foregoing, this gear ratio corresponds to reverse gear and is therefore a negative figure. If the number of revolutions in the reverse gear is called -k, (30) may be written as

$$1 : -k \text{ (reverse)} \qquad (31)$$

Upon regulation to the opposite extreme position of the gear (as shown in fig. 2), the gear ratio passes through neutral gear to forward gear. A similar calculation of the maximum gear ratio in this extreme position can now

be made. Such a calculation will show that the maximum gear ratio for the forward gear will be k + 1, i.e. the gear ratio will be

$$1 : k + 1 \text{ (forward)} \qquad (32)$$

To the forward gear ratio one revolution of the input shaft is to be added when the output shaft, in case of forward as well as reverse, is supposed to make one revolution, because the ball rings always rotate oppositely the rotary direction of the input shaft and thus gives the reverse gear the improved gearing range.

Generalization of the geometry of the transmission mechanism

On the basis of an analysis, which will not be repeated here, of the capacity of the transmission mechanism of transmitting power and torque it can be shown that the conditions mentioned in the foregoing do not allow optimum use of the permissible surface pressure between the balls when the quantities n, R_k, α and c are selected so as to provide a great gearing width. If optimum use of the permissible surface pressure between the balls and a great gearing width are to be obtained at the same time, the condition that the circular arc-shaped shifting paths of the ball centres intersect each other on the major axis of the imaginary ellipse, as shown in fig. 12, must be abandoned. The reason is that optimum use of the permissible surface pressure occurs at such a great value of the angle α that the balls in one ball ring touch each other in the marginal gearing position, as shown in figs. 28 and 29. However, if a great α value is selected and the shifting paths of the ball centres intersect each other on the major axis of the ellipse, the width of the gearing range will be limited because the inward

shifting of the balls must not entail that their points of tangency with the rolling paths exceed the x' axis.

However, the condition relating to the intersection of the ball centre paths just represents a borderline case, where the distance between the major axis of the imaginary ellipse and one extremity of the shifting path of the ball centres is equal to zero. This distance, called m, may, however, be between zero and the distance marked x" in fig. 30. Thus, if a suitable value of the quantity m is allowed, optimum use of the surface pressure between the balls and great gearing width can be obtained at the same time.

In a sense it is possible to calculate a gear in which the present ball gear principle is fully utilized both in respect of forces and gearing width. However, this requires that the calculation procedure is changed on one point more. The addition of the margin m does not guarantee that the quantity c is selected so that the greatest gearing width occurs at optimum use of the surface pressure between the balls. To ensure this, the formulae must be amended so that the ball point of tangency with the moulding ring at the marginal gearing position lies in the intersection of the x' axis, as shown in fig. 31. This provides the greatest variation of the involved radii and thus the greatest gearing width.

However, these two improvements of the geometry are possible only in theory. In practice, it cannot be justified either that the balls in one and the same ball ring touch each other, or that the ball points of tangency with the moulding rings lie in the x' axis; but the closer this theoretical ideal can be approached in practice, the better.

Both improvements are determined by the axial shifting of the balls, so the interblocking of the balls as well as the transgression of the actual extent of the rolling paths can be prevented by adding a security margin t on the axial shifting c, as shown in fig. 32.

In the event that a greater gearing width is desired than the one made possible by the above-mentioned amendments of the calculation procedure, a smaller value of the angle α than calculated is selected. But an increase in the gearing width obtained in this manner will be at the expense of the optimum use of the surface pressure between the balls.

Variants

The geometrical principle in the regulation of the ball gear can be applied in several different ways. Four such variants are schematically shown in figs. 33-36. However, these do not represent any improvement in relation to the arrangement of the rolling paths where they are symmetrical in pairs and have balls of the same size in both ball rings, but just changes in the characteristic relating to the regulation of the gear ratios.

The individual variants of the gear can, in addition to the use on the ten different types, be varied in a special manner, the aim of which is to shift the gearing range. Figs. 37 and 38 schematically show such a special arrangement of a gear type 14-3 in one and the other extreme position, respectively. As will be seen from the figures, neutral gear, like before, is obtained when the ball centres are disposed in the x' axis; but owing to the different circular arc-shaped cross-sections of the ring tracks neutral is here in one marginal gearing position. On the other hand, the regulation extends

further to the other side so that an increase in the gearing width to that side corresponds to the restriction to the other. In other words, the gearing range can in this manner be shifted to one or the other side of the gear ratio scale. Thus, different ball gear types can be manufactured, each of which covers various sections of the gear ratio scale.

Patent Claims

1. A transmission mechanism containing two sets of balls which form co-axial rings and are so supported by a pair of inner and a pair of outer ring-shaped rolling paths co-axial with the ball rings that each ball in one ring touches two balls in the other ring, and vice versa, and that the ball rings can rotate about the common main axis upon relative rotation of the rolling paths, and each ball can additionally rotate about its own individual axis through its centre, the inner and outer rolling paths being axially shiftable with respect to each other within predetermined limits, characterized in that the axial distance between the inner as well as the outer rolling paths is constant.

2. A transmission mechanism according to claim 1, characterized in that the rolling paths have such concave-curved generatrices that, in all relative positions of the inner and outer rolling paths within said limits, the individual axes of rotation of the balls intersect the corresponding outer and inner rolling paths or imaginary, inwardly extending extensions thereof at points within the points of tangency of the balls with the rolling path in question, and that the distances of each outer or inner point of tangency from the main axis and the individual axes of rotation vary oppositely upon relative shifting of the rolling path pairs, and that the ratio of these distances is always greater for the outer rolling paths than for the inner ones.

3. A transmission mechanism according to claim 2, characterized in that the rolling paths have substantially circular arc-shaped generatrices.

4. A transmission mechanism according to claim 2 or 3, in which the balls in the two sets are of the same size, characterized in that each pair of opposite rolling paths has symmetrical generatrices.

5. A transmission mechanism according to claim 4, characterized in that the four rolling paths have double-symmetrical generatrices.

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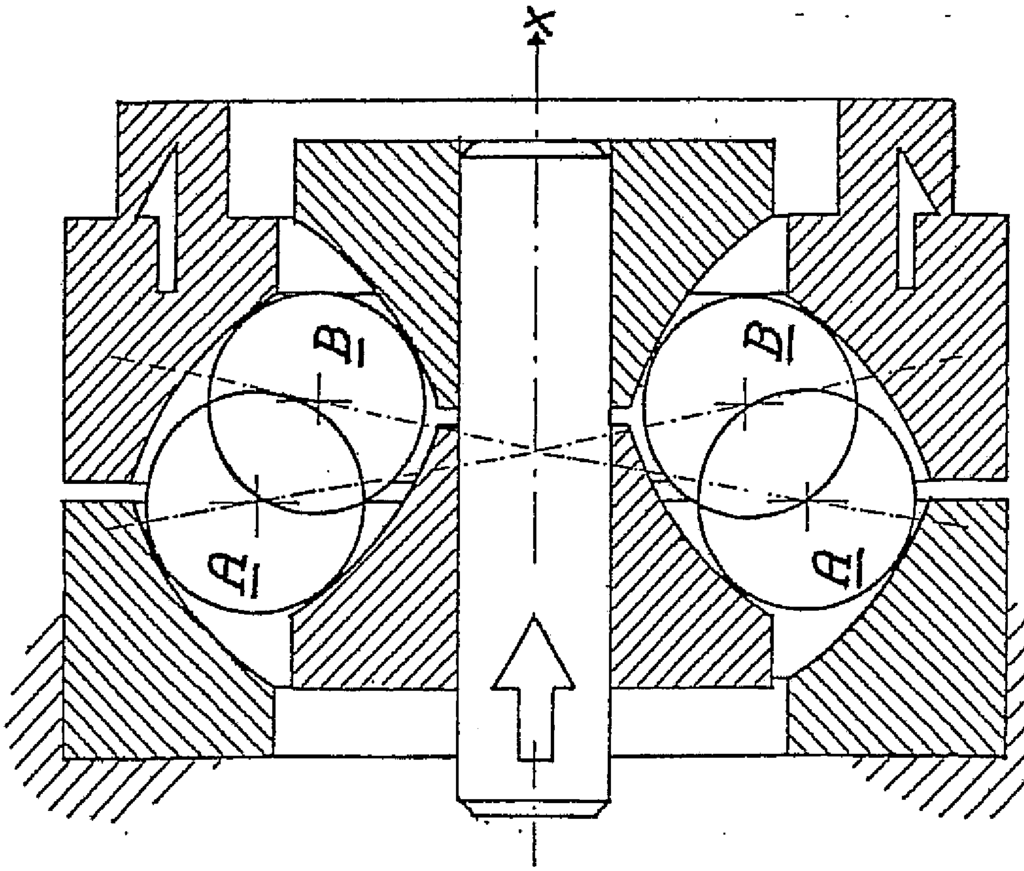

Fig.2

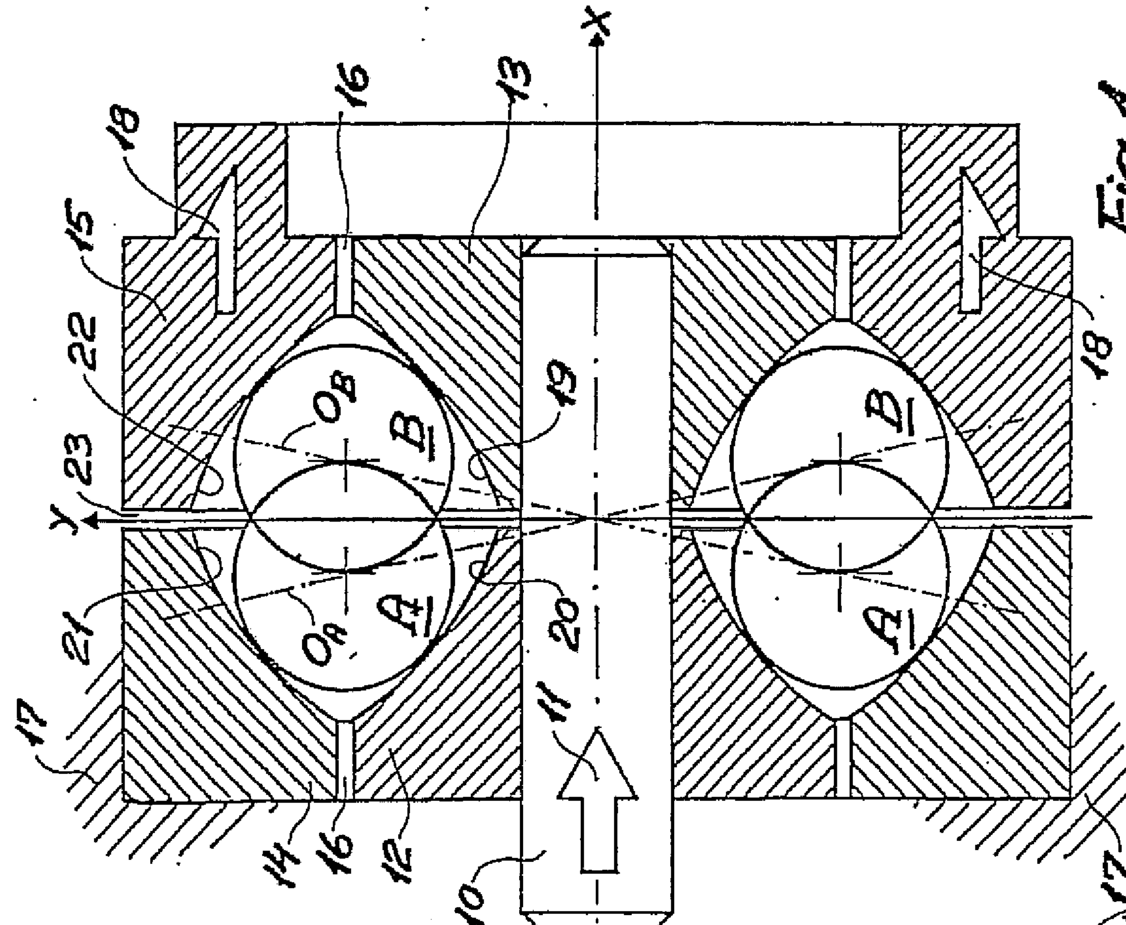

Fig.1

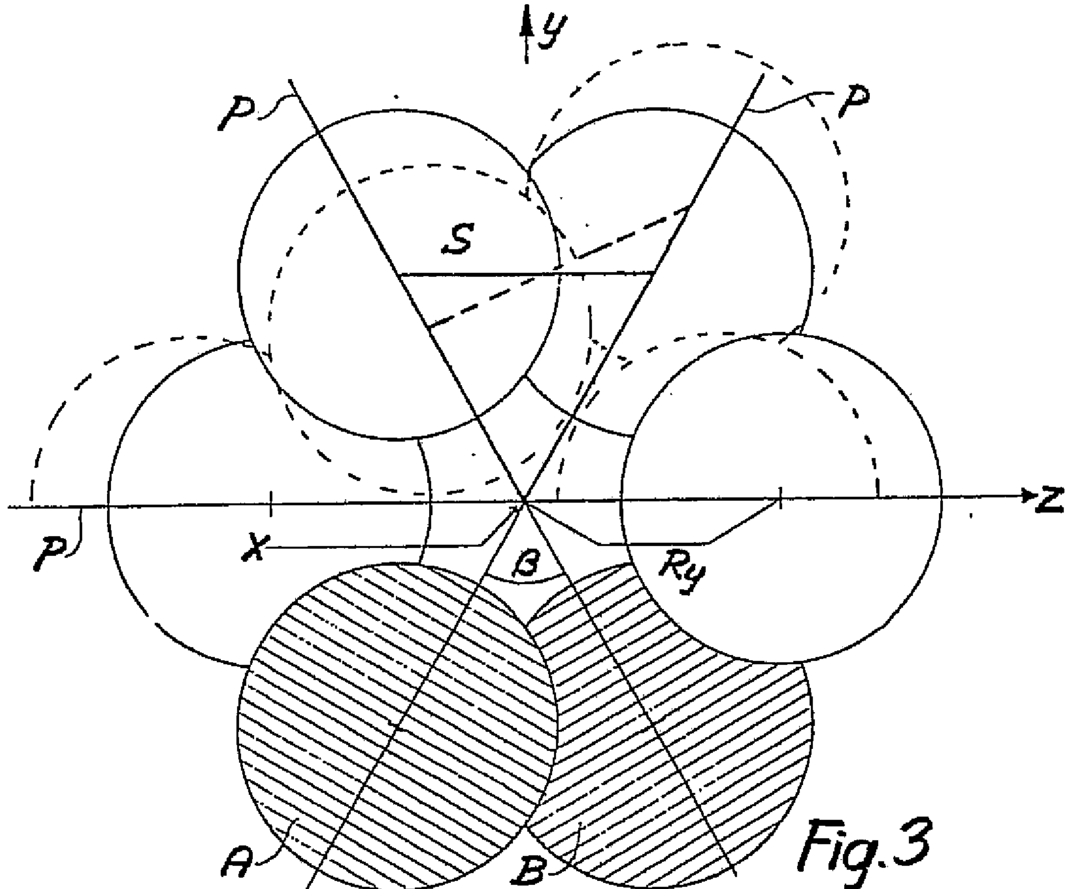

Fig.3

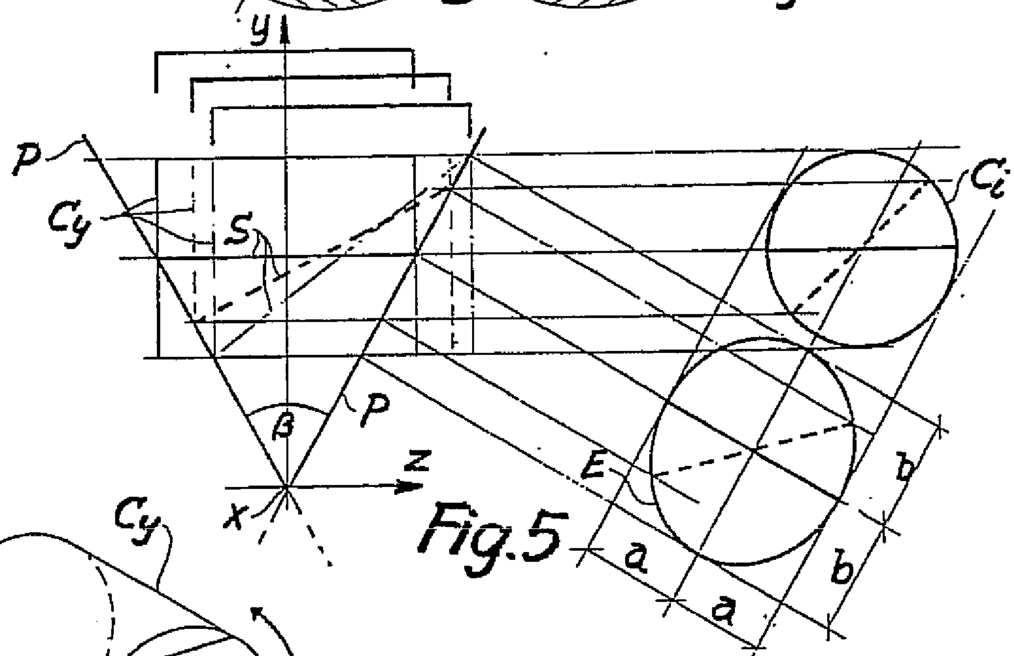

Fig.5

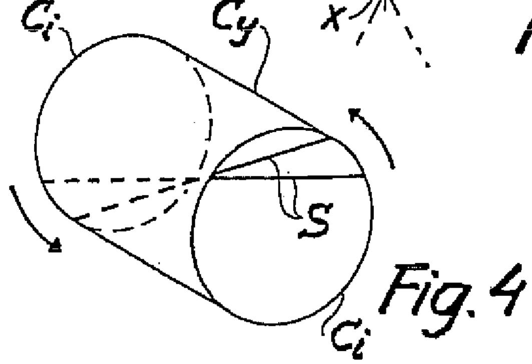

Fig.4

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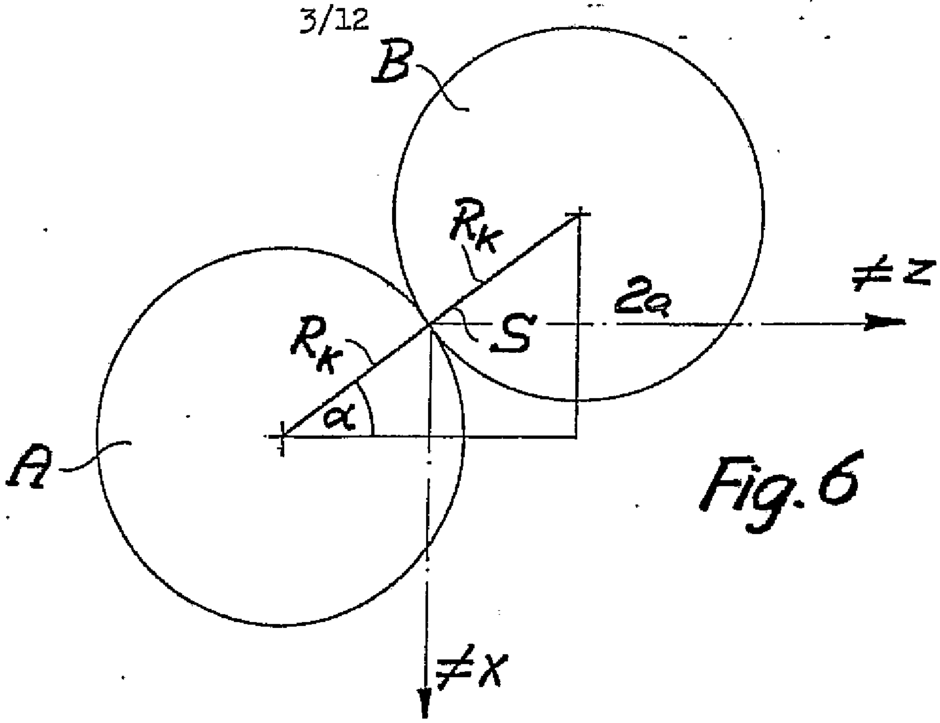


Fig.6

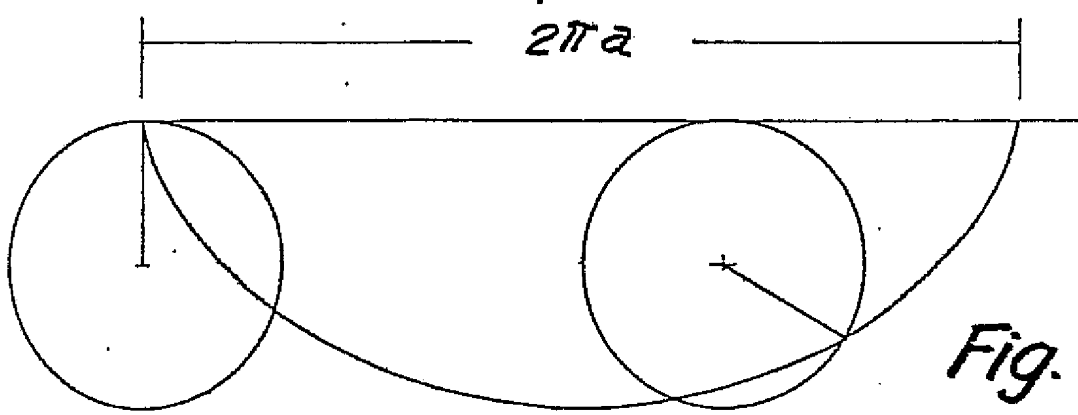


Fig.7

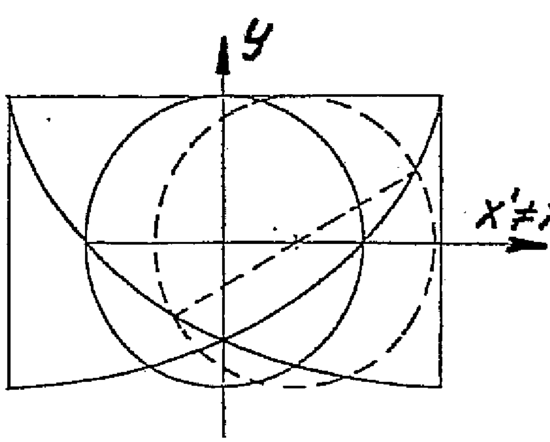


Fig.8

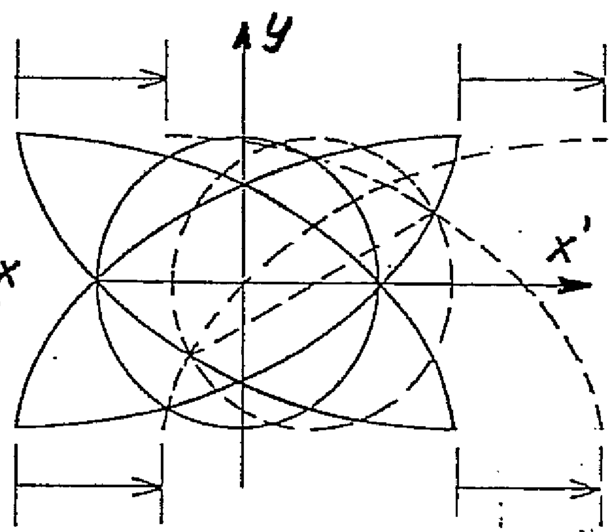


Fig.9

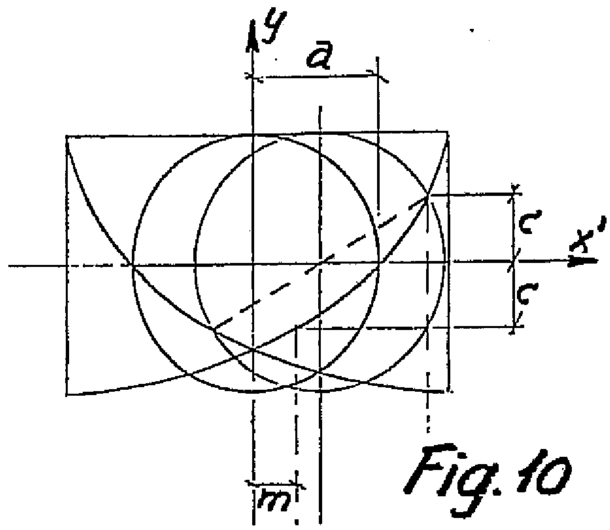


Fig. 10

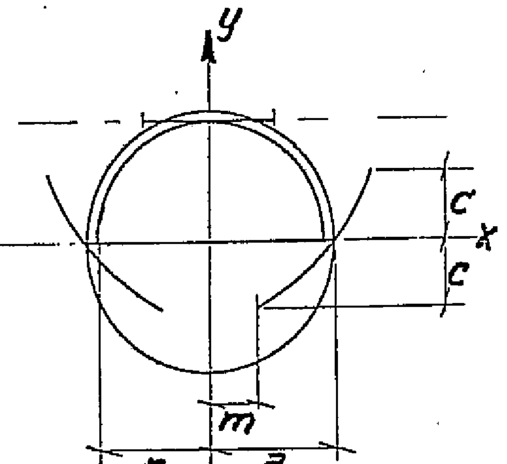


Fig. 11

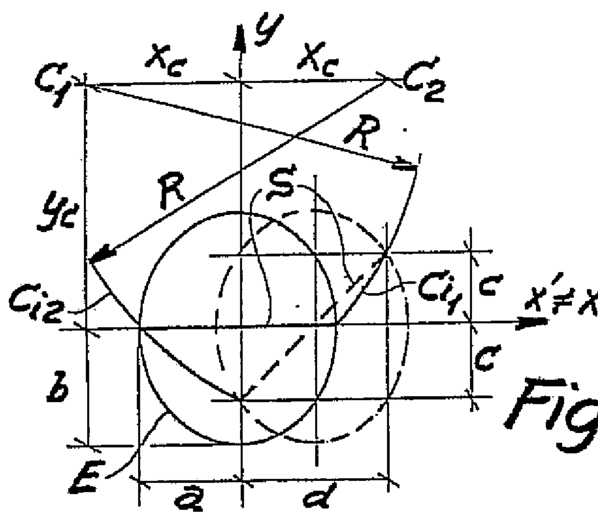


Fig. 12

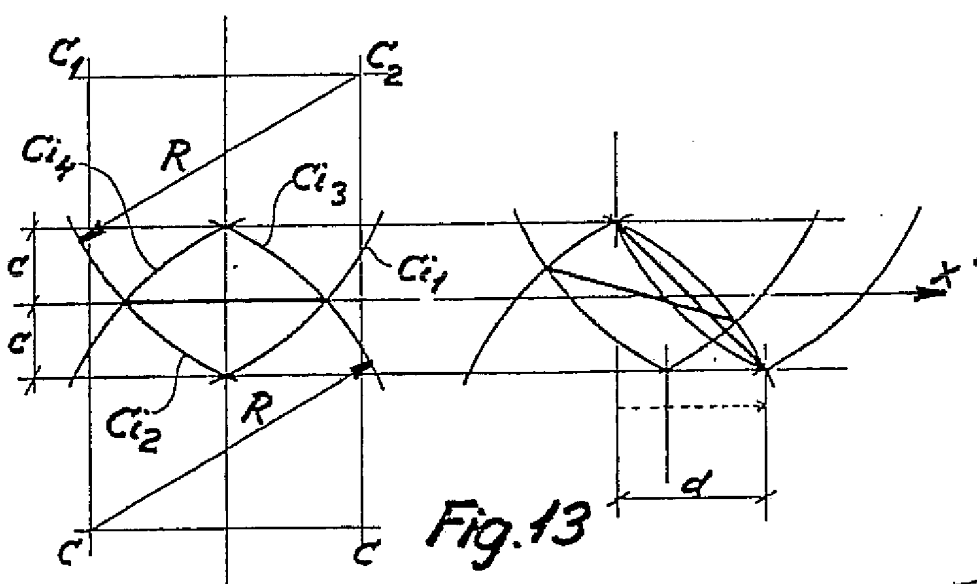


Fig. 13

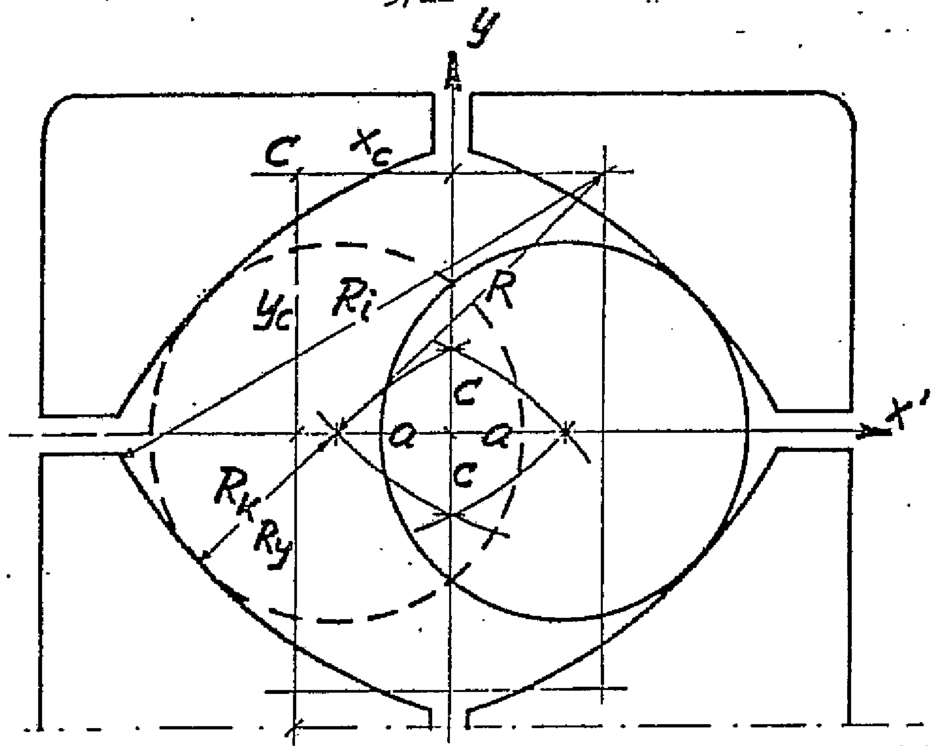


Fig. 14

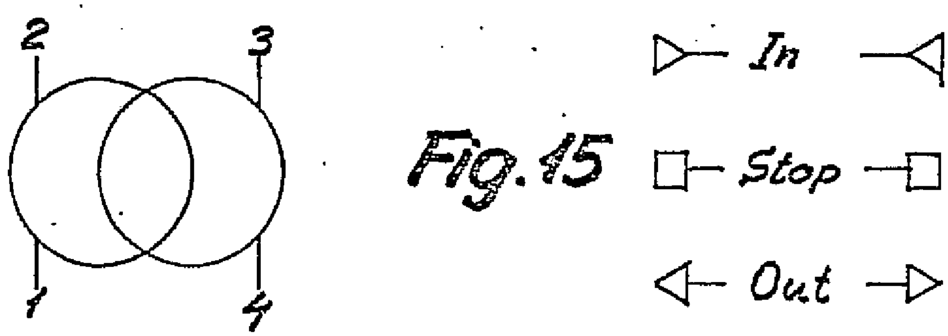


Fig. 15

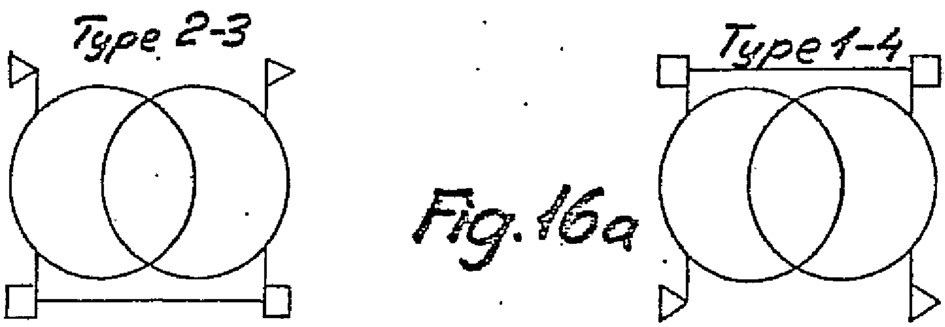
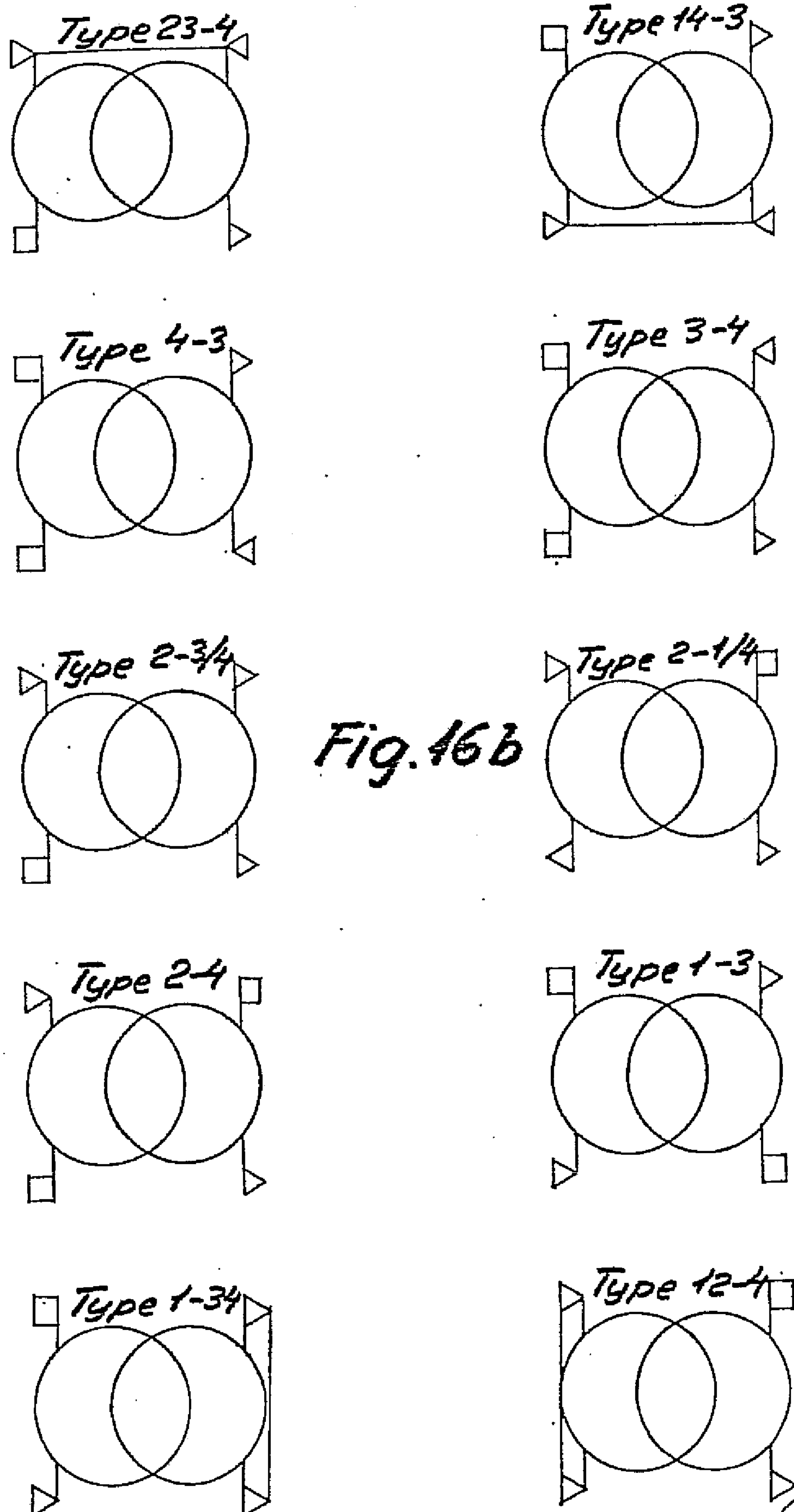


Fig. 16a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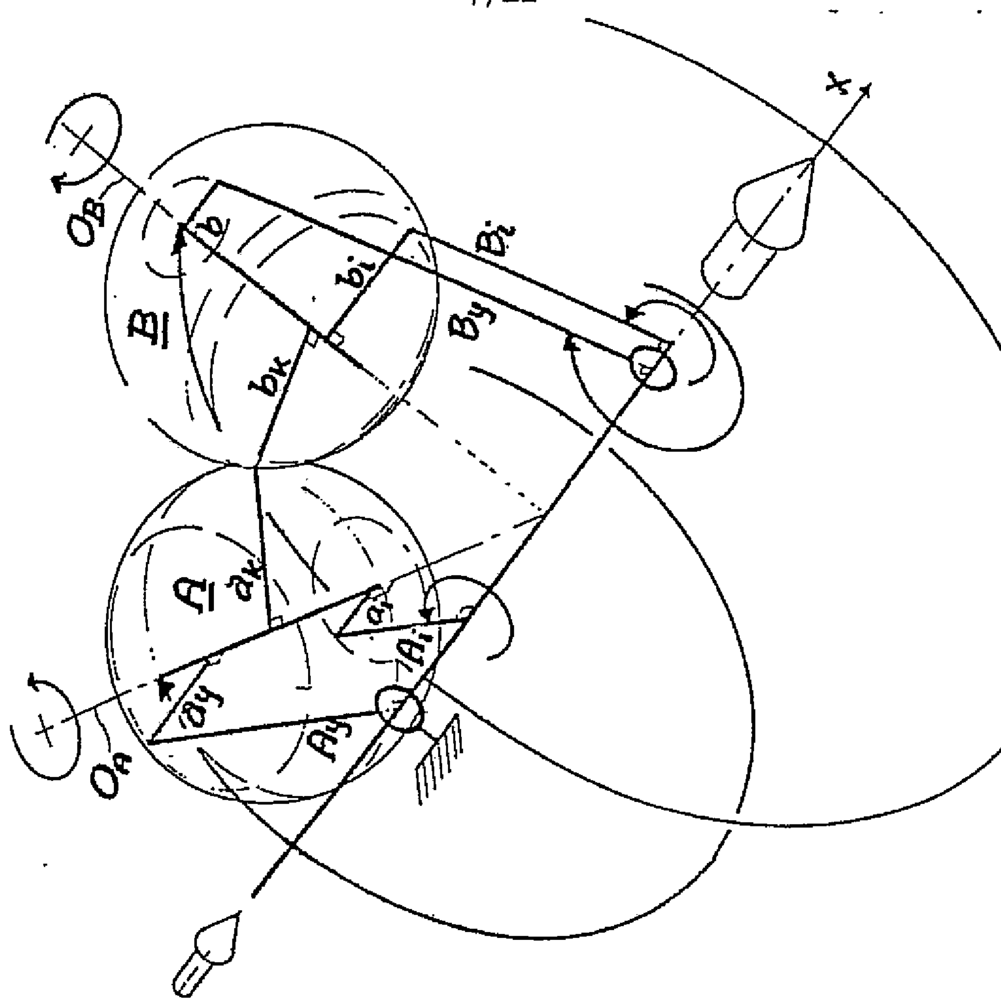
Type 3-4
Type 2-3/4
Type 2-1/4
Type 2-4
Type 1-3
Type 1-34
Type 12-4

Fig. 16b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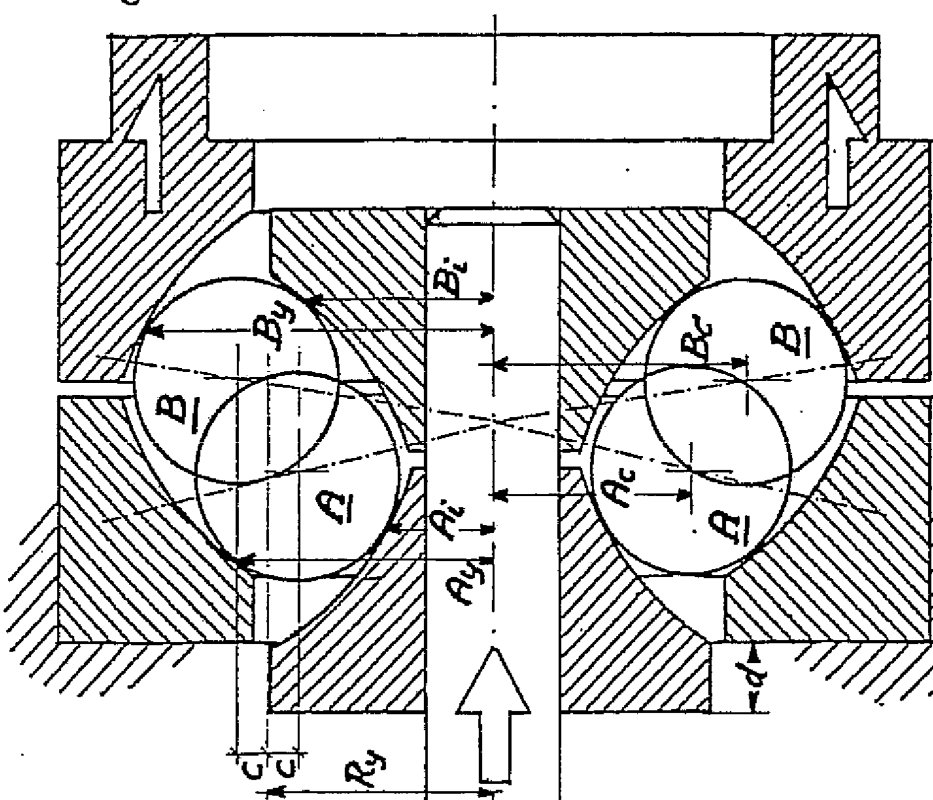
$\underline{B}$
b
b_i
B_i
B_y
b_k
a_k
$\underline{A}$
a_i
A_i
a_y
A_y
O_A

Fig. 18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B_y
$\underline{B}$
B_c
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A_i
A_c
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A_y
d
c
c
R_y

Fig. 17

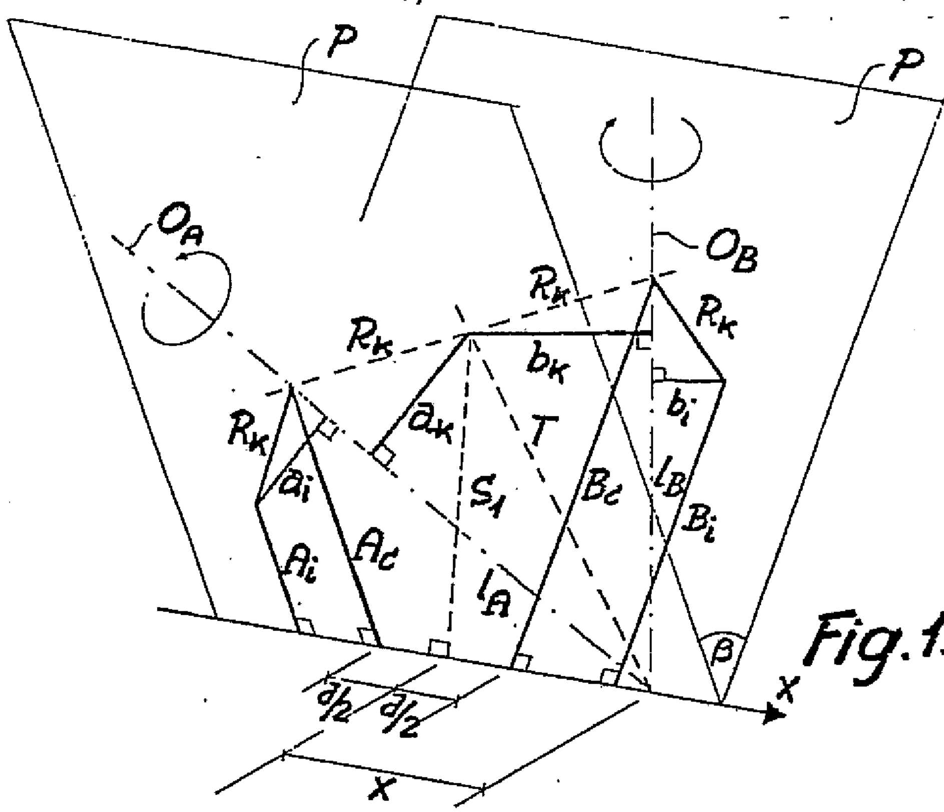

Fig.19

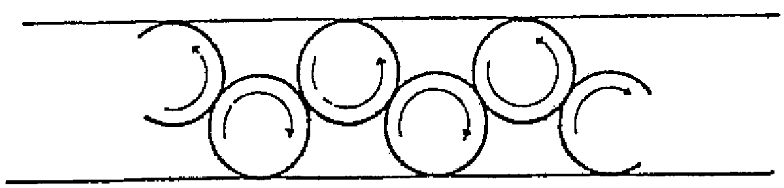

Fig.20

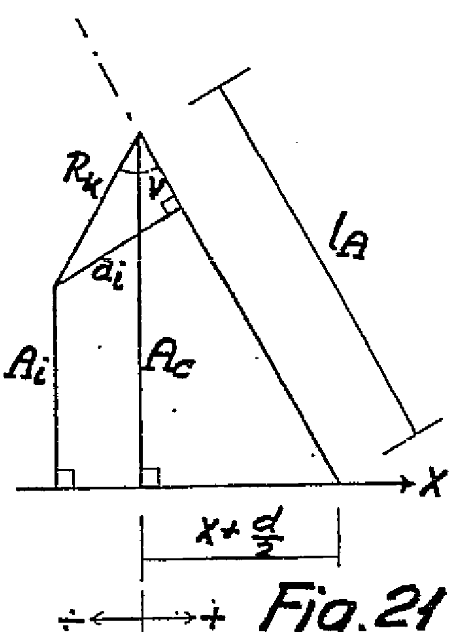

Fig.21

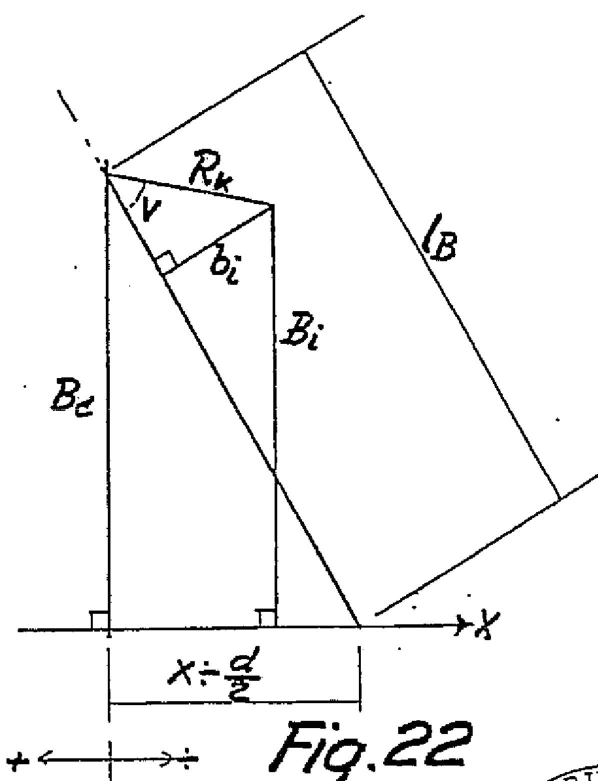

Fig.22

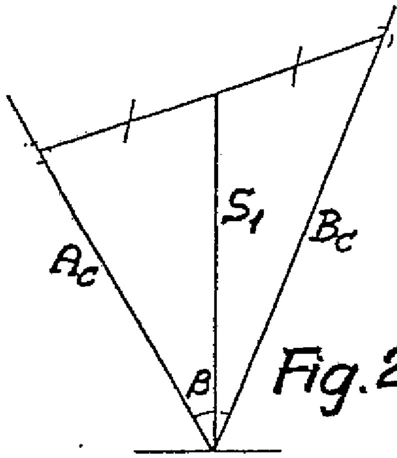

Fig.23

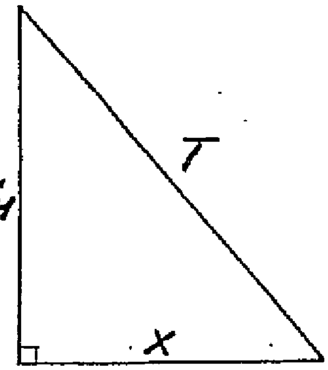

Fig.24

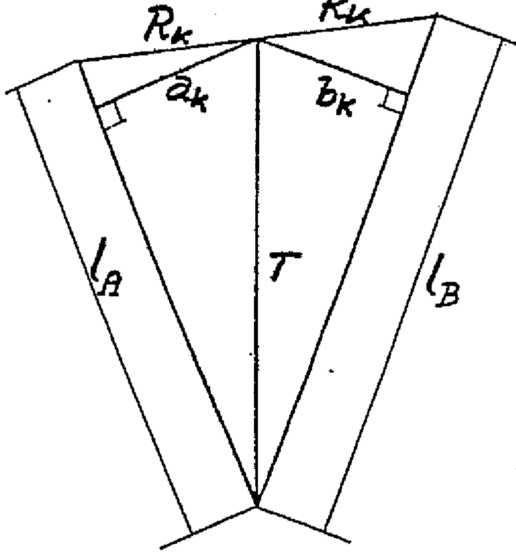

Fig.25

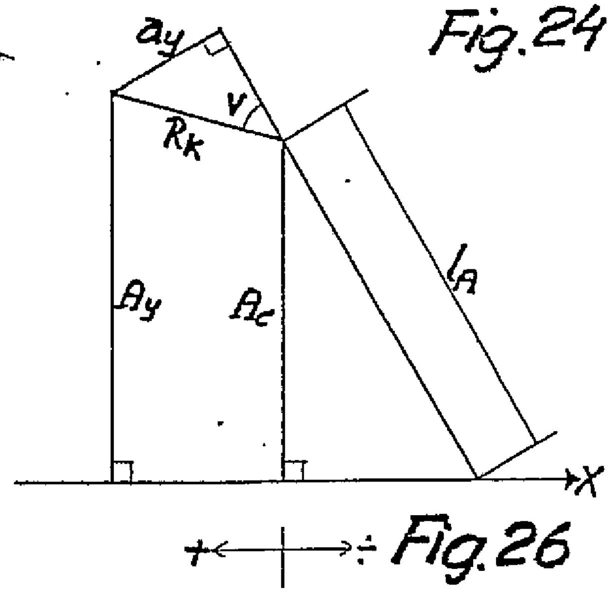

Fig.26

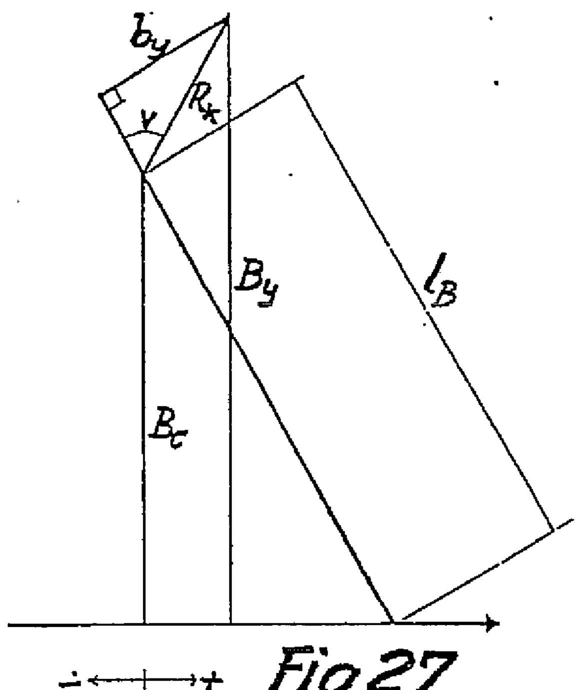

Fig.27

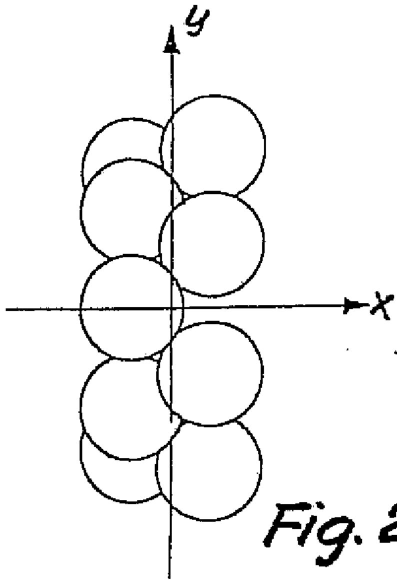

Fig. 28

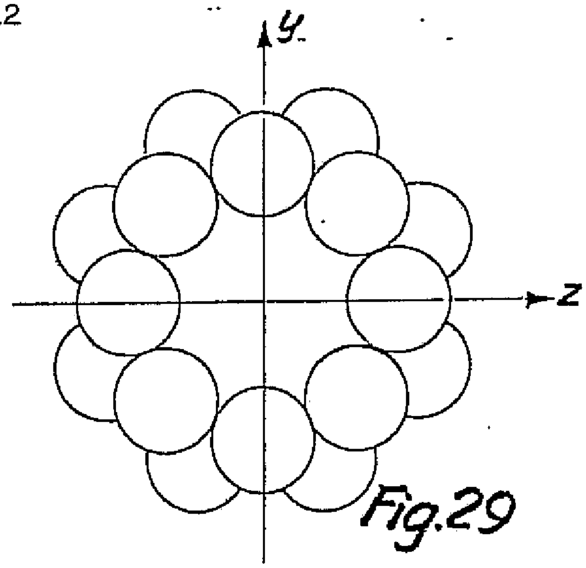

Fig.29

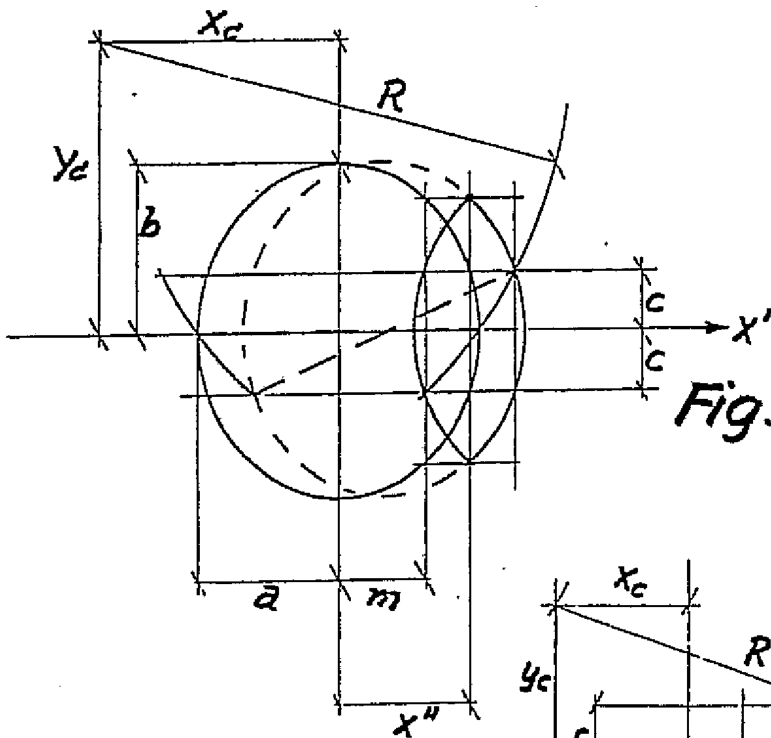

Fig.30

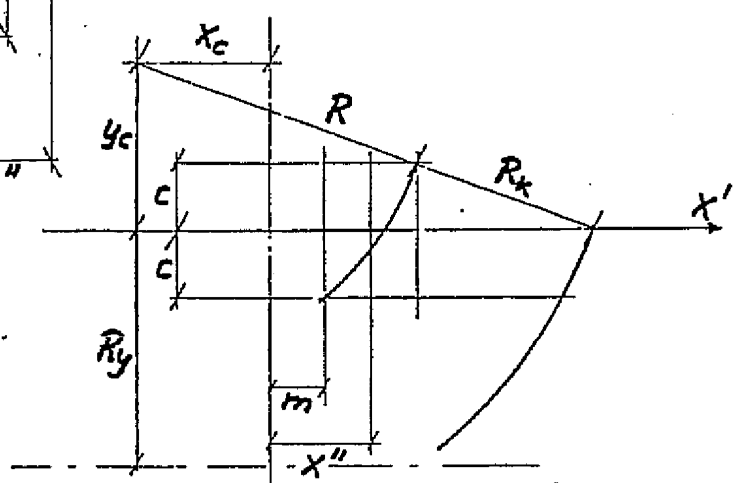

Fig.31

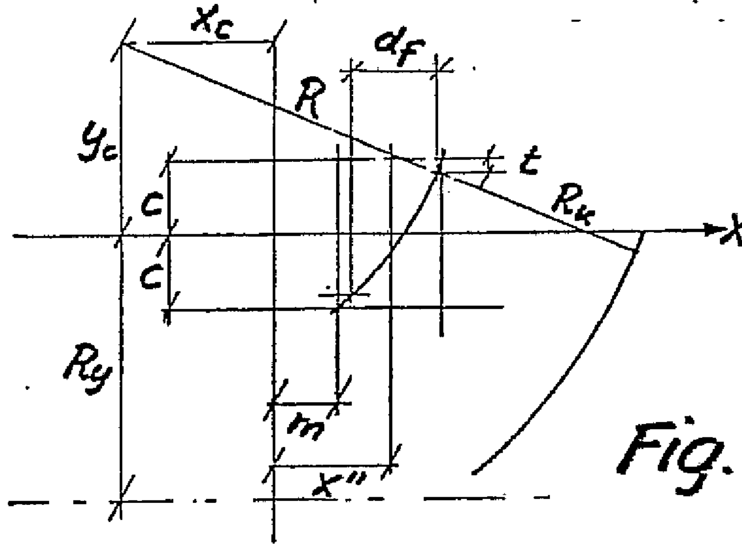

Fig. 32

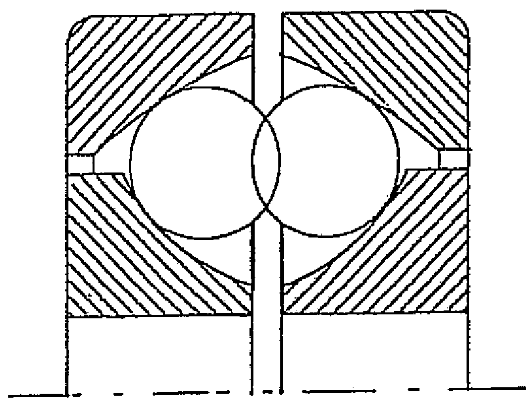

Fig. 33

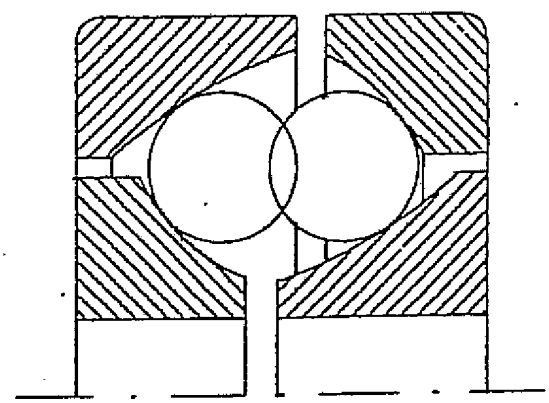

Fig. 34

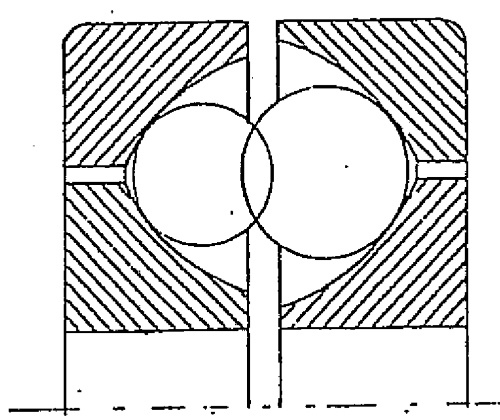

Fig. 35

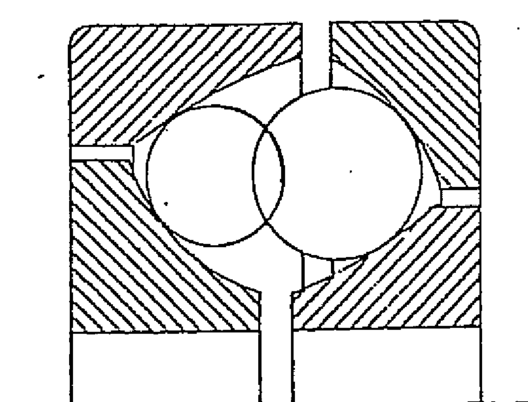

Fig. 36

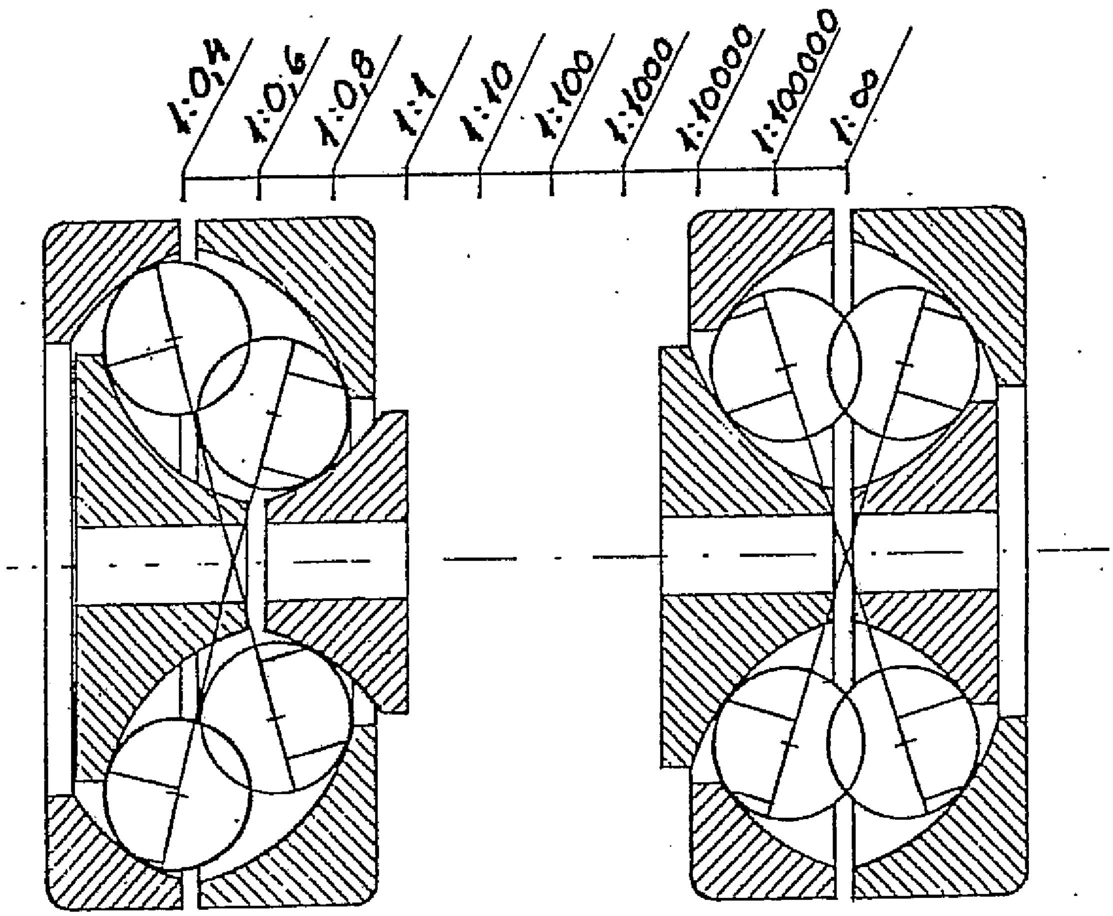

Fig. 37

Fig. 38

INTERNATIONAL SEARCH REPORT

International Application No PCT/DK83/00016

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) [3]

According to International Patent Classification (IPC) or to both National Classification and IPC 3

F 16 H 15/50

II. FIELDS SEARCHED

Minimum Documentation Searched [4]

Classification System	Classification Symbols
IPC 3	F 16 H 15/40, 15/48-15/54
National Cl	47h:14
US Cl	74:198, 793, 796

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched [5]

SE, NO, DK, FI classes as above

III. DOCUMENTS CONSIDERED TO BE RELEVANT [14]

Category *	Citation of Document, [16] with indication, where appropriate, of the relevant passages [17]	Relevant to Claim No. [18]
X	SE, C, 151 580 (LUTZ O) 20 September 1955 see fig 2, 4	1 - 4
X	US, A, 2 862 407 (LUTZ O) 2 December 1958	1 - 4
X	US, A, 2 878 692 (WOLF M) 24 March 1959	1 - 4
X	DE, C, 877 082 (LUTZ O) 2 April 1953	1 - 4
X	DE, C, 926 887 (LUTZ O) 31 March 1955	1 - 4
X	DE, C, 901 852 (LUTZ O) 26 November 1953	1 - 4
X	DE, C, 911 083 (LUTZ O) 1 April 1954	1 - 4
X	GB, A, 739 917 (FIAT SOC) 2 November 1955	1 - 4
A	GB, B, 798 263 (ELLIOTT BROS) 16 July 1958	

* Special categories of cited documents: [15]

"A" document defining the general state of the art which is not considered to be of particular relevance

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"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

IV. CERTIFICATION

Date of the Actual Completion of the International Search [2]	Date of Mailing of this International Search Report [2]
1983-03-31	1983-04-27
International Searching Authority [1]	**Signature of Authorized Officer** [10]
Swedish Patent Office	Artur Emtedal

Form PCT/ISA/210 (second sheet) (October 1981)